MACHINE DESIGN

January 1953, Vol. 25—No. 1

THE PROFESSIONAL JOURNAL FOR ENGINEERS AND DESIGNERS

Over the Board

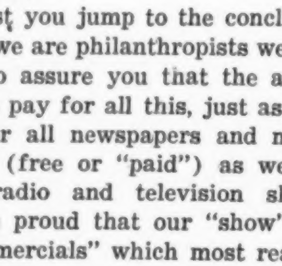
What Does It Cost?

Ever try to figure out what it might be costing to place this copy of MACHINE DESIGN in your hands? Based on last year's cost figures, the answer is well over three dollars per issue, not counting provision for taxes—almost two dollars per pound. Of course we realize that the value to you is many times that figure, but as a cost-conscious engineer you may be interested in the fact that our nominal subscription price of $10 per year (83 cents per copy), which we do not charge design engineers, would not even pay the mechanical costs of typesetting, engraving, printing, paper, and ink.

Lest you jump to the conclusion that we are philanthropists we hasten to assure you that the advertisers pay for all this, just as they do for all newspapers and magazines (free or "paid") as well as for radio and television shows. We're proud that our "show" has "commercials" which most readers find just as welcome as the "program" itself.

This Month's Cover

In the communication of ideas, television is simply following in the footsteps of engineers who have long depended primarily on pictures to transmit ideas, information and instructions. But the standard orthographic "blueprint" leaves much to be desired, and the recent trend has been toward a better picture, just as TV engineers keep striving for better tubes and circuits to improve their picture. Thus John Lane's article on "Perspective Drawing" on Page 106 has such timely significance that we asked Penton artist George Farnsworth to design a front cover to highlight the subject. The values of such pictorial presentation extend far beyond the conventional production illustration with which it is generally associated.



Holes, Big and Little

Some months ago we drew attention to the specific meaning now attached to the word "shake," which is defined as one one-hundredth of a millionth of a second. Now we find that so general a word as "minute" (small) has specific dimensional significance, at least in relation to hole sizes. A recent court decision has ruled that any hole of 0.050-inch diameter or less is a minute hole, which will be news to many engineers with somewhat different ideas on relative size. For another aspect of holes, turn to Page 116 in this issue.

Control System Stability

We are told that all control systems fall into one of two categories—open loop and closed loop. A third type has come to our attention but so far no name has been proposed for it. It seems that Mr. and Mrs. acquired a dual-control electric blanket for their double bed. Somehow the controls were transposed so that Mr. controls the Mrs. side and vice versa. When Mrs. feels chilly she turns up her control, making Mr. too warm, so Mr. turns down his control making Mrs. even chillier. What happens after a few cycles of this is best left to the imagination.

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Published on the seventh of each month. Subscription in the United States and possessions, Canada, Cuba, Mexico, Central and South America: One year $10. Single copies, $1.00. Other countries one year, $15. Acceptance under Act of June 5, 1934. Authorized July 20, 1934.

Cadillacs or Hot Rods?

A WELL-KNOWN aircraft designer has observed that our military machines may be likened to Cadillacs in contrast to the "hot-rod" quality of Communist equipment. The parallel is not too far-fetched. A military aircraft is strictly a utility vehicle and the concept of luxury should not enter the picture.

What seems to have happened is this: those responsible for the design of military aircraft have very properly considered the overall effectiveness of the fighting machine plus its human crew, and have proceeded on the premise that whatever contributes to the comfort and safety of the operators automatically improves the performance of the combination.

The result is a degree of complexity and cost which is assuming extravagant proportions and may defeat the whole purpose of the approach. Each unit of added weight on aircraft requires bigger wings and engines, more fuel, heavier frame and landing gear, and so forth, creating a vicious spiral that leads to a tenfold weight increase overall. The counterpart of Cadillac's automatic headlight dimmers, pushbutton window lifts, air conditioning, etc., on a fighter plane can therefore lead to fantastic weight and cost increases. A piece of equipment weighing as little as one hundred pounds adds half a ton to the weight of the plane and, at current fighter plane prices, forty thousand dollars to the cost.

Implications of the situation so dramatically exemplified by fighter plane design go far beyond the confines of this particular type of machine. The present business boom is largely artificial and undoubtedly will soon give way to a more solidly based economic situation in which spending will be much less free and buyers of all sorts of goods, including industrial equipment and domestic appliances, will be critical of nonessentials.

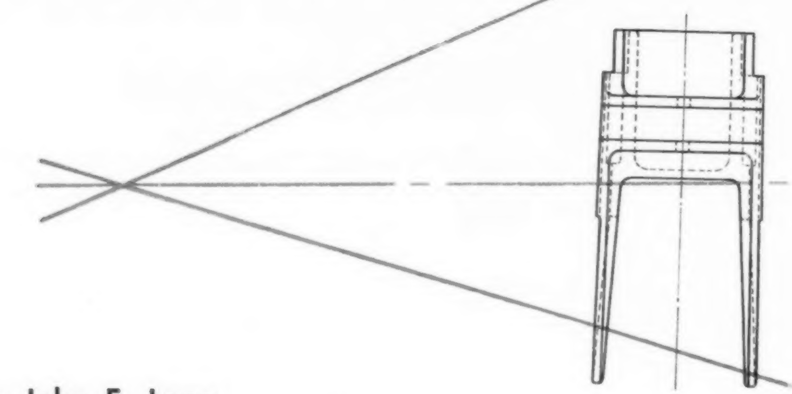
Now is the time for design engineers to give the most earnest attention to developing utilitarian designs against the days when customers are less able to afford frills. Careful analysis of the overall effectiveness of any dispensable feature in terms of the man-machine combination is a must. A little less Cadillac and a little more jeep in our design philosophy would be a healthy trend.

Colin Carmichael
EDITOR

Engineering

PERSPECTIVE DRAWING

Presenting engineering information in three dimensions



By John F. Lane

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North American Aviation Inc.
Downey, Calif.

ENABLING the designer to present engineering information in clear, easily understood form, three-dimensional illustrations have already gained wide acceptance in several fields and have striking possibilities in supplementing or replacing conventional three-view drawings.

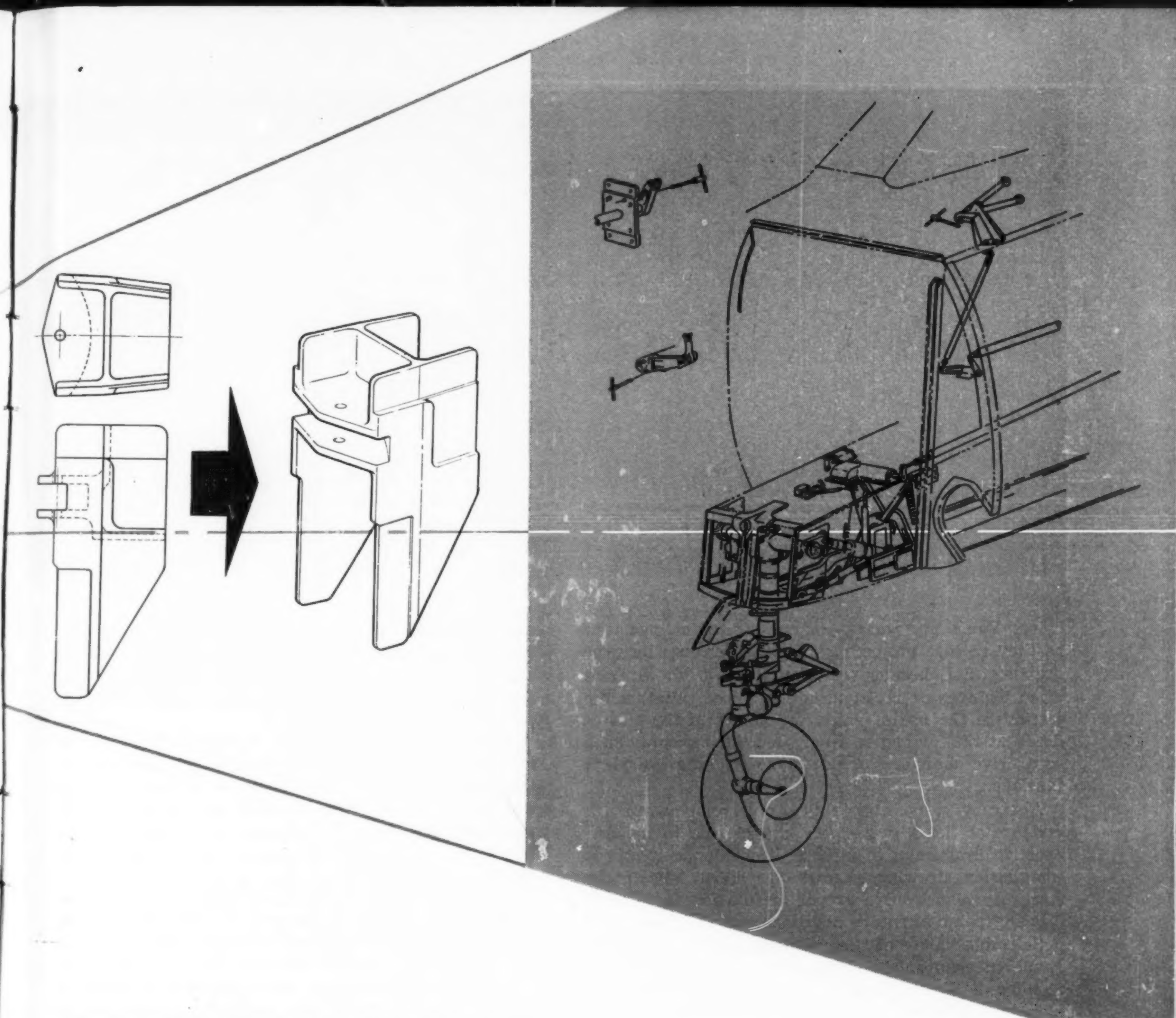
Among the many advantages of perspective engineering drawings is the lucid visualization possible and the simplified portrayal of design information, especially to untrained personnel. During the war, for example, with the huge influx of less-skilled workers into the aircraft industry, the need for basic improvements in presentation of engineering data became increasingly apparent.

At first, a few illustrations were used to supplement ordinary orthographic, or three-view, drawings in more complex structures. These were so successful, however, that in many job operations it was decided to completely replace the orthographic presentation. Such three-dimensional illustrations enable relatively unskilled workmen to visualize clearly what the designer had in mind. Moreover, in structures whose cross sections change rapidly along a given axis, a single perspective drawing replaces a great number of the complex and unnatural orthographic drawings which actually interfere with attempts to visualize a part in its finished form.

In 1943, a form of drafting was introduced at North American Aviation, Inc. which at once provided the advantages of the standard orthographic drawing and of the full-scale mock-up of the subject. The method was called "engineering perspective drafting" and was greeted with considerable enthusiasm by the manufacturing departments of the company, including tooling, subassembly, final assembly, cost estimating, purchasing, and material control. The method has continued in use and through evolution it has gained wider application. At present, it is used almost exclusively for the presentation of certain types of design information to the manufacturing shops.

Engineering perspectives are true production drawings, carrying such information as the bill of materials and call-outs for parts and equipment. They are begun as soon as the shape and size of the subject have been established. They are drawn accurately to scale, and from them structural elements can be laid out as the design progresses. Thus, the various systems can be worked out on these drawings almost as easily as in a mock-up.

Uses of Perspective Drawings: Need for some form of pictorial representation is greatest in the presentation of engineering information to the shop. The worker's output is greatly increased, in some cases as much as 50 per cent, when illustrations are used to portray machine operations, sequencing, and assembling details.



During critical periods when skilled labor is not readily found, industry is forced to fill the gaps with novice workers. Many of these people have no prior experience with engineering blueprints. Not only does the use of engineering perspectives reduce the training period of such people, but is also cuts down reading time and reduces costly shop errors which usually occur when inexperienced personnel attempt to interpret complicated blueprints.

Occasionally, in more complicated engineering design layouts, the designer may not be able to picture the item exactly as it would appear when fabricated. A perspective illustration worked in with the preliminary design enables the designer to accomplish his job more effectively and also allows him to inject the aesthetic values sometimes needed.

Wide use of perspectives simplifies and expedites the presentation of information in important consultations, both in the shop and in engineering. In engineering, some of the most useful pieces of technical art are perspective inboards, cutaways, and exploded views. These give a clear picture of the total efforts on any project. Other uses for the persective inboard and exploded views, supplemented by detailed break-down illustrations, are in the production-planning department for developing assembly line techniques, and in the tooling department for working out preliminary tooling designs.

Perspective drawings give prospective bidders and customers a clear understanding of proposals and related information by presenting all features and details in brief and concise fashion. Also, management finds illustrated technical data almost indispensable in top-level briefing and in personnel-training programs.

Production Perspective Drafting: The perspective system is a form of axonometric drawing, using parallel lines and true direct measurements. This style was selected rather than the true perspective method, using vanishing points, because of its simplicity and adaptability to the standard engineering drafting ma-

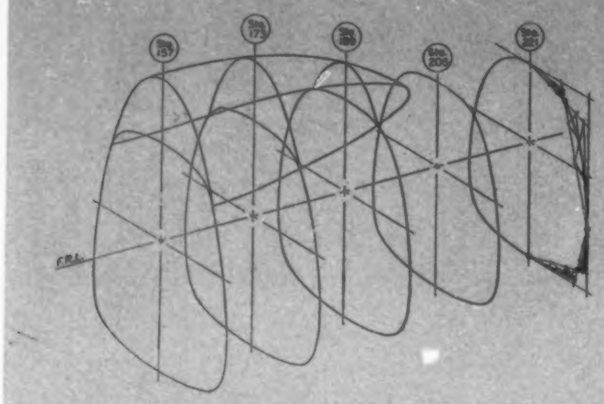


Fig. 1—Perspective drawing of master lines gives basic lines for all later drawings

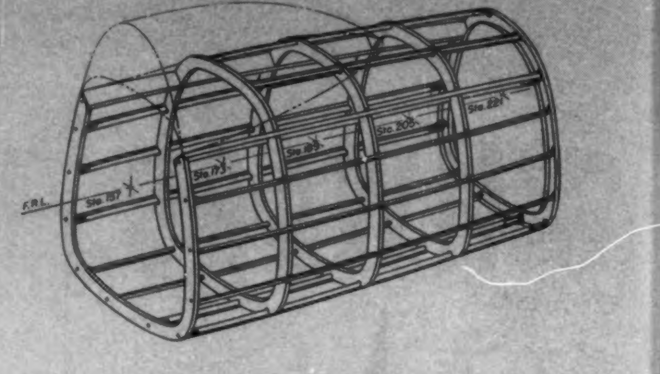


Fig. 2—Layout of structure becomes the master structure drawing for making installation drawings

chine. Vanishing points are not desirable, inasmuch as they require projecting all measurements into the drawing, and because re-orientation is so difficult when changes or revisions are required. Another objection to the use of vanishing points in the loss of detail at the far end of a long, slender assembly such as a drive shaft or the sideways view of an airplane fuselage.

Accuracy in measurements is not important for the end use of the drawings but is a big aid to the perspective draftsman in structural development and in installation drawings showing systems and parts. Accuracy also insures proper orientation for a work starting point in future additions and changes.

Selecting values of the angles for the working axes should depend solely upon the product in design. A shallow angle lends itself to long, slender assemblies, since it fits them into the standard roll-size drawing. Deeper angles may be considered for shorter assemblies, that is, where the three major dimensions are of the same magnitude. Another consideration in selecting the working angles is the desired viewing level of the layout: the steeper the working angles, the higher the viewing level.

Working scales of perspective drawings are generally determined by the size of the subject, the extent of the breakdown into sub-assemblies, and the drawing size. The scale range may vary from full size to one-eighth size. In aircraft work, where the assembly breakdown is quite extensive and comparatively small sections are illustrated on a single drawing, the usual scale is about one-fourth. This large scale permits greater detail in installation drawings and reduces the necessity for supplementary enlarged views.

An important point to remember in the use of this system is its flexibility which, in turn, allows the perspective draftsman considerable freedom in his layout of the drawing.

Drawing Development: The steps in development of production perspective drawings for installation and assembly are as follows. Although an airplane structure (airframe) is used as an example in these steps because it is sufficiently complex to bring out the features of the perspective system, the system is applicable to any major asembly.

Axis Angles and Scales: Selecting the angle for each axis and the drawing scale is very important. Care must be exercised because the whole drawing is affected by these values. Consideration must be given to the purpose of the drawing, the desired viewing angle, the shape and complexity of the object, and the location of significant components.

Basic Lines Layout: With dimensional information taken from a dimensions book or from loft boards, master lines are developed for body stations, wings and tail and are given appropriate station numbers as shown in *Fig.* 1. Accuracy at this phase governs the accuracy obtainable in the remaining phases of the drawing.

Development of Structure: Based on preliminary structural engineering layouts, layout of the structure for each subassembly is carried out, preferably on an overlay vellum, thus keeping the basic lines drawing free for future use. This layout, *Fig.* 2, consists of frames, bulkheads, stringers and longerons and becomes the master structure drawing from which installation drawings for the various parts of the system are made.

Finished Installation Drawing: The master structural drawing may be used as the basis for any drawings to show location of components of a particular system. Separate drawings are made for each of the systems; thus, the master structural drawing is used anew for each system, such as electrical, hydraulics, instrumentation and fuel. A vellum is laid on the master drawing and the desired system components are spotted over the structure. Information for the parts and equipment to be installed is obtained from the design installation layout; at times the information is taken directly from a mock-up or an experimental model. Just enough structure is traced onto the drawing to tie in the installation completely

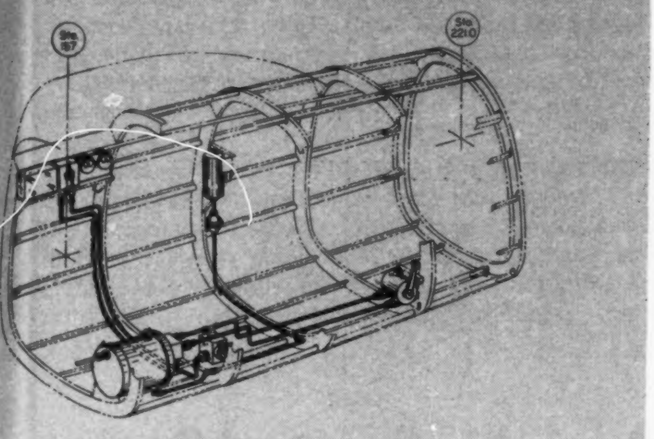


Fig. 3—Finished installation drawing shows location of system components in master structure

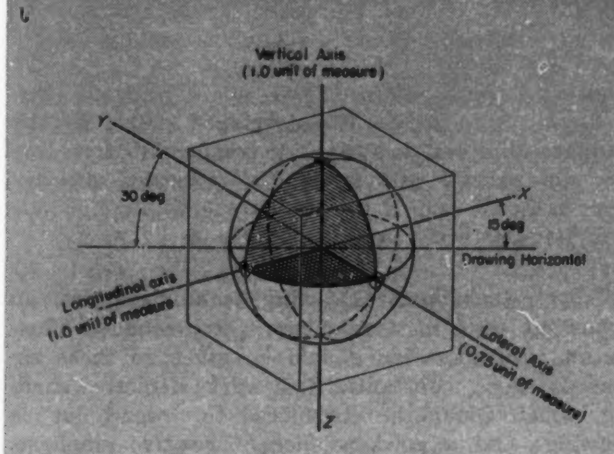


Fig. 4—Basic working axes of the perspective drawing

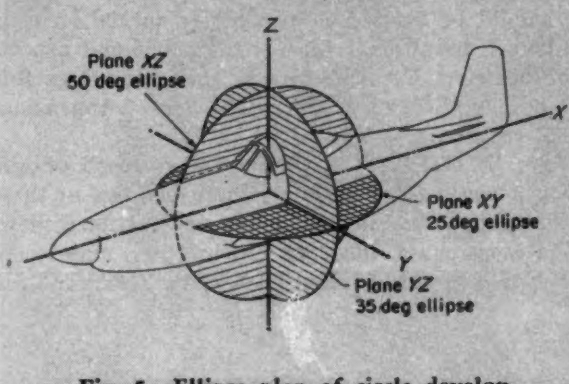


Fig. 5—Ellipse plan of circle development for aircraft perspective drawing

and to orient the user in the shop with respect to the actual assembly. An installation drawing, such as *Fig.* 3, is completed with the addition of the necessary detail auxiliary views, call-outs and the bill of materials.

A Working System: Embodying the principles explained above, a working system has been developed and proved successful at North American for the preparation of engineering drawings for guided missiles and related control equipment.

The basic working axes are selected to give a pleasing eye view with sufficient depth to permit the showing of a reasonable amount of top plane information. Schematic representations of the axis layout are shown in *Figs.* 4 and 5. Axis *Z*, the drawing vertical represents the vertical of the ship. A true dimension is used in measuring along this axis and all lines parallel to it. The longitudinal or along-ship axis *X* is laid out at a 15 degree angle from a drawing horizontal and requires use of true dimensions. Line *Y* at a 30 degree angle from the horizontal is used for the lateral, or across-ship axis. An accurate axis development shows that measurements along this axis and all lines parallel to it should be shortened ¼-inch to the inch. This foreshortening maintains a mechanical development correctness that is needed in this system, and also partially compensates for the apparent distortion inherent in regular axonometric projections.

Ellipse patterns of circle development for the three working planes, which are determined by the angles of the axes, are shown in *Fig.* 5. The horizontal plane *XY* and all parallel planes require a 25 degree ellipse. The vertical lateral plane *YZ* and its parallel planes take a 35 degree ellipse. The vertical longitudinal plane *XZ* and all parallel planes require a 50 degree ellipse. All circle developments that lie askew of the working planes are proportionally larger or smaller-degree ellipses, depending upon position.

Measurements must always be made along lines parallel to the working axes. A measurement to be taken, or a line to be located, on the off angle requires use of the technique of offset measuring. *Fig.* 6 shows a typical case of locating a line that lies askew of the working axis, Point *A*, which is on the longitudinal axis, is located by measuring x_1 directly along the axis from some convenient station or reference point. Point *B* is located by three offset measurements, x_2, y and z, which are all parallel to the working axes. Engineering design layouts provide the dimensional information needed in measuring these offsets.

Development of basic outlines which are composed of compound curves is accomplished in the same manner as in orthographic layouts. Two commonly used systems for this are the second-degree curve system, and the standard co-ordinate system. Ordinates for the second-degree system are first located in perspective. Information required in this system consists of centerline points, maximum half-breadth, and shoulder points. From these, the curve may be developed by

PERSPECTIVE DRAWING

connecting these points, *Fig.* 7, and laying out lines 1, 2 and 3 to locate the developed point. In the standard co-ordinate system of compound curve layout, dimensional information is measured directly from a layout and is located on the water and buttock plane network as shown in *Fig.* 8.

Proper Techniques: Drawing techniques play an important part in developing a well-rounded illustration. Consideration should be given to these essentials: good, substantial line work; depictive shading, which should be considered to "round out" a drawing; and a good balance of relative emphasis between reference structure and installation material to be specified. Other than ordinary drawing equipment, principal tools required in preparing a perspective drawing are a drafting machine, ellipse guides, and a foreshortened ruler to make direct measurements along the lateral axis.

If dimensions are required on perspective drawings, they should be limited to over-all or special locating dimensions, because of the three-dimensional aspects of the drawing. For this reason, these drawings find their best application when multitudes of dimensions are not required.

It can be seen that with the added depth obtainable in a perspective drawing, the usual two or three orthographic views and the necessary sectional views are, in most cases, incorporated into one main view. Time is thus saved by elimination of the need for shuf-

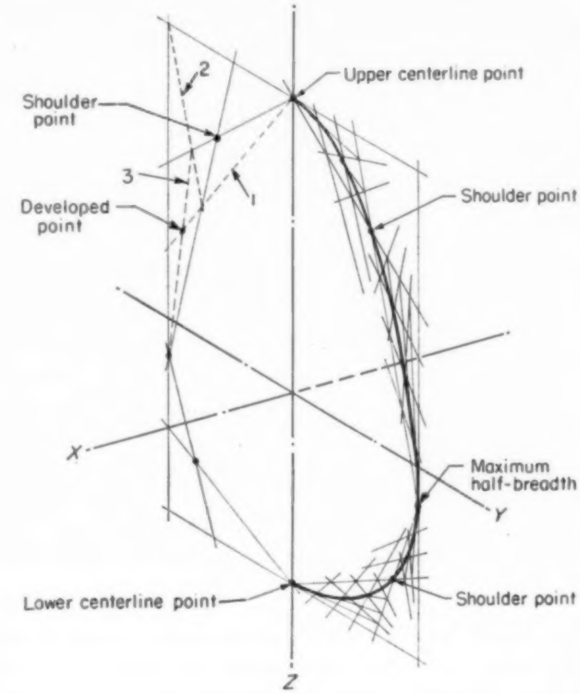
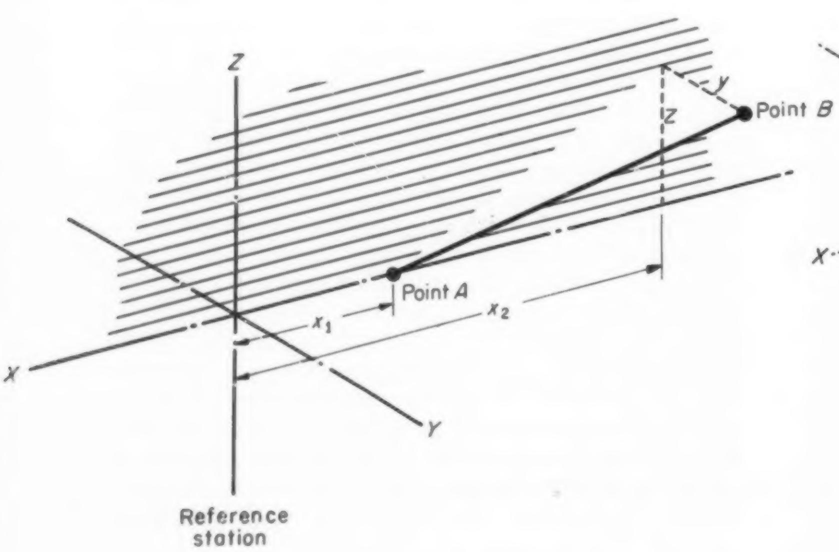
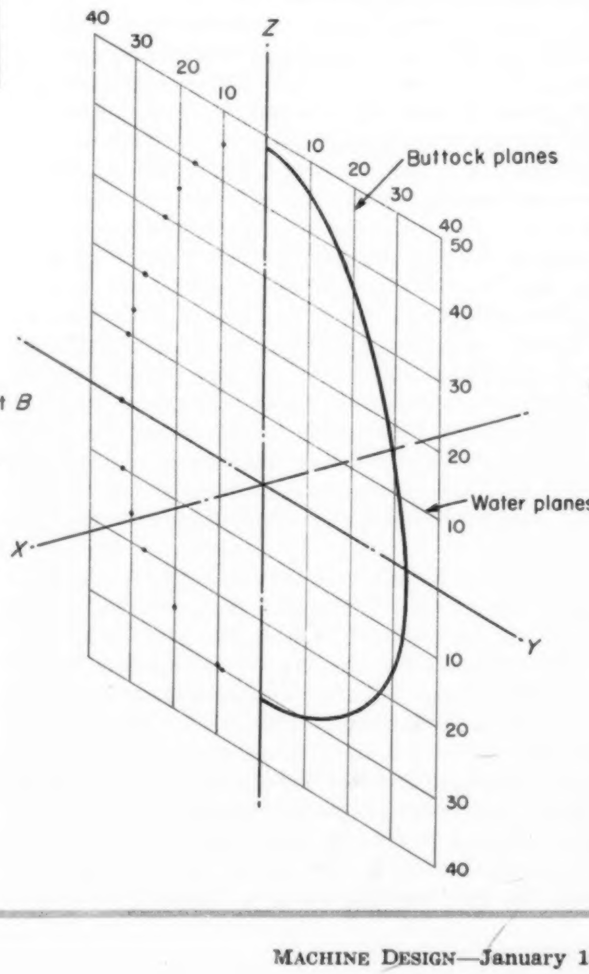


Fig. 7 — Above — Development of a compound curve by the second-degree curve system

Fig. 8—Below—Standard coordinate system for laying out a compound curve

Fig. 6—Location of a line on an off angle





fling through long, multifold blueprints and attempting mentally to tie together information contained in different views. It will be found also that the preparation time for perspective drawings needs to be no greater than for the orthographic drawings to be replaced.

Summary: Use of perspectives can, in many different ways, effect savings in time and money. These include: reducing print reading time; reducing break-in time for new employees; eliminating many shop errors; aiding development techniques for tooling and production; aiding management briefings; implementing training programs and field service work.

Because of the clear and concise presentation of data, further savings are possible because the same drawings can either be used "as is" or can become the basis for the artwork necessary in the written material required on a job. Such artwork is useful in process specifications, factory check-out procedures, instruction manuals, product catalogs, parts manuals, and sales brochures.

It can be seen that the uses of perspective drawings are practically unlimited. As the system is developed in a particular field, new applications, limited only by designer's imagination, will be found.

Plastic Used in Building Scale Models

EXPERIMENTAL work at the David Taylor Model Basin with scale models made of plastic and glass cloth indicates that this combination of materials has several advantages over wood and paraffin, which have been employed in models. Plastic models are lighter and easier to handle than other types and are equally as strong, factors which are particularly important in high velocity wind tunnel tests. Also, plastic models can be ballasted more accurately, retain accurate dimensions longer, and the thinner shells leave more space inside the models for the installation of instruments.

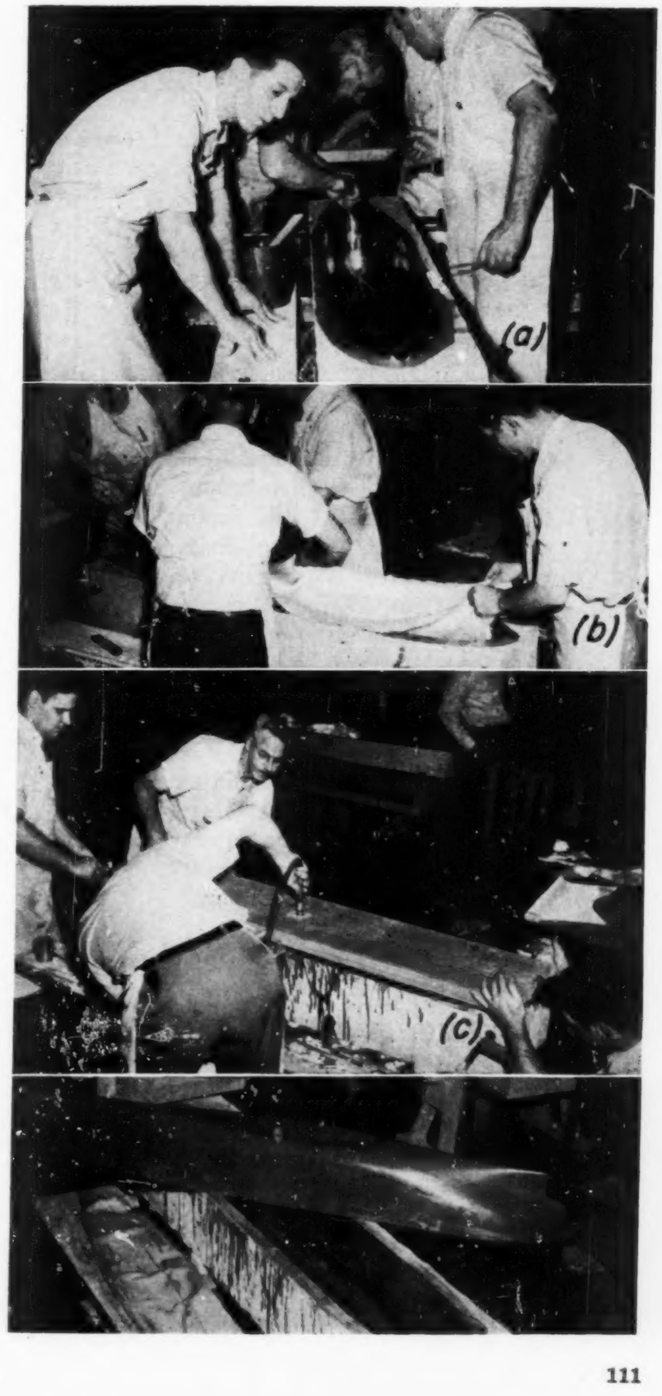
One method for constructing such models consists of making a wood or wax pattern which is finished with lacquer, coated with wax, and polished to facilitate its release from the mold. The pattern is then coated either with an adhesive or a mixture of equal parts of tin and bismuth sprayed to a thickness of 1/32 to ⅛-inch. It is next suspended in a box filled with plaster, which serves as a backing to the adhesive or metal.

The pattern is removed and liquid latex is sprayed or brushed into the mold to a thickness of approximately 1/32-inch. A board large enough to cover the plaster mold is sprayed or brushed with latex and cemented to the latex in the mold to form a balloon. An air valve is inserted in the board.

Surface of the mold is coated with polyvinyl alcohol dissolved in water, a silicone release and a plastic binder to which color has been added, *a*. Layers of glass cloth 0.003 to 0.015-inch thick are fitted to the mold, *b*; each layer is coated with the plastic binder until a shell from 1/32 to ¼-inch thick has been formed.

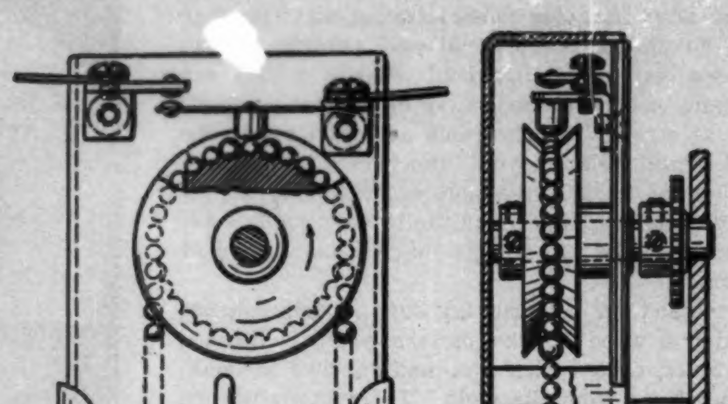
The deflated latex balloon is placed in the mold again, clamped securely and subjected to air or water pressure of 10 to 30 pounds for 24 hours, *c*. After the model is removed from the mold, *d*, and polished, transparent plastic decks and cover plates are sometimes added. Appendages to the model can be installed easily.

The mold and balloon can be used many times if numerous identical models are needed, and alterations of the original pattern as well as the mold can be made easily.



SCANNING the Field for IDEAS

SELECTIVE ELECTRICAL PROGRAMMING is obtained through the use of endless bead chain in this timer developed by M. A. Blouin of Simplex Time Recorder Co. Circulating through a divided storage compartment and over an intermittently driven sprocket, the chain serves much like a pattern chain by being fitted with clips or jackets at the desired intervals. Passage of the enlarged beads over the sprocket actuates a switch in the signal circuit. Chain length is virtually unlimited and individual beads can be selected to represent minutes, half minutes or any other desirable interval. Program alterations are readily made by rearranging the spacing of the oversize beads.



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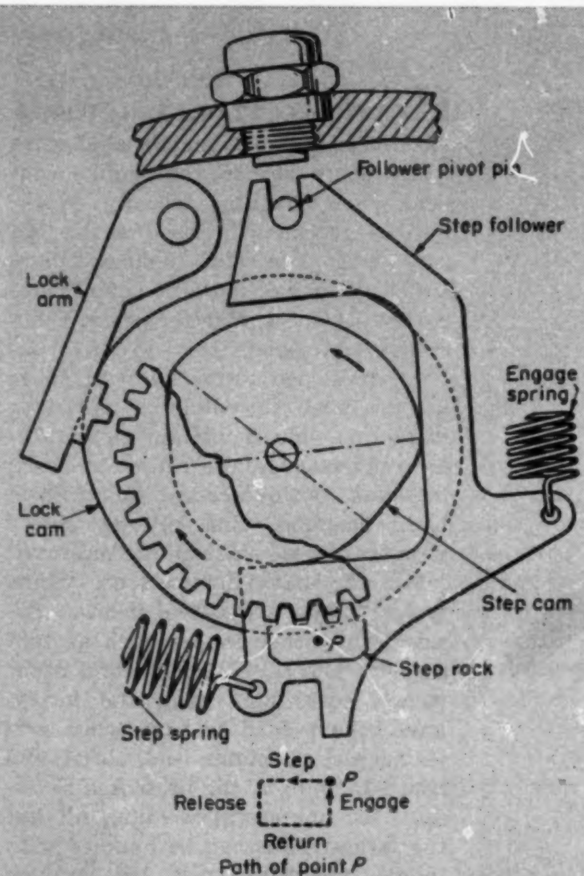
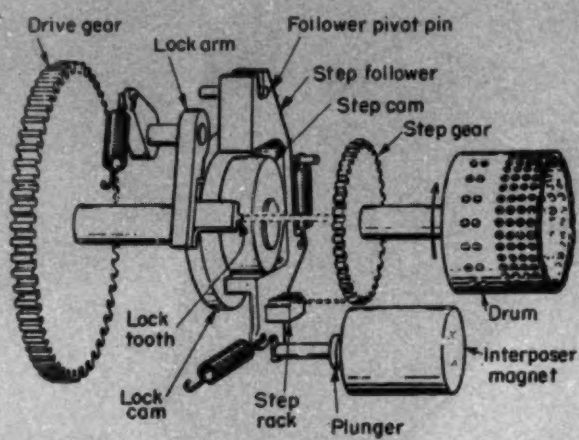
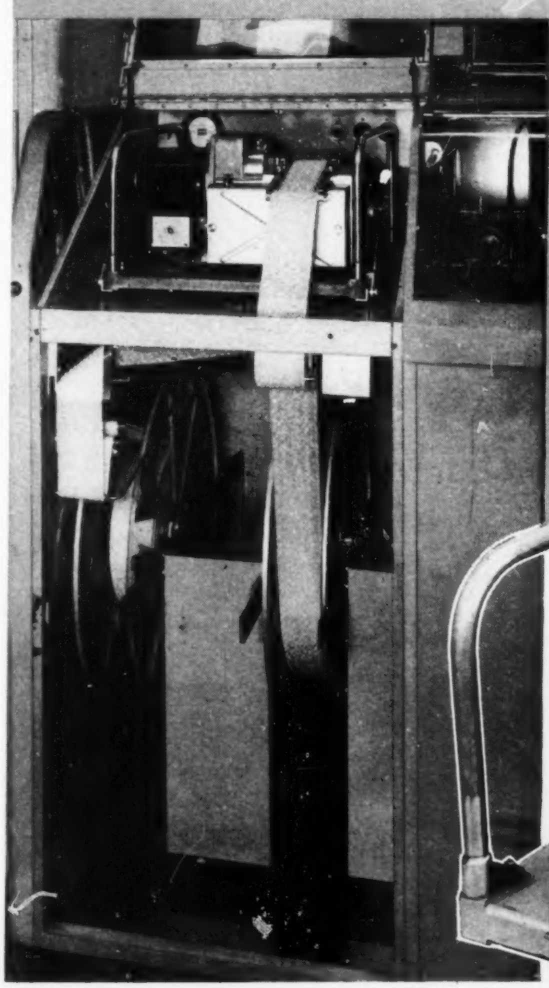
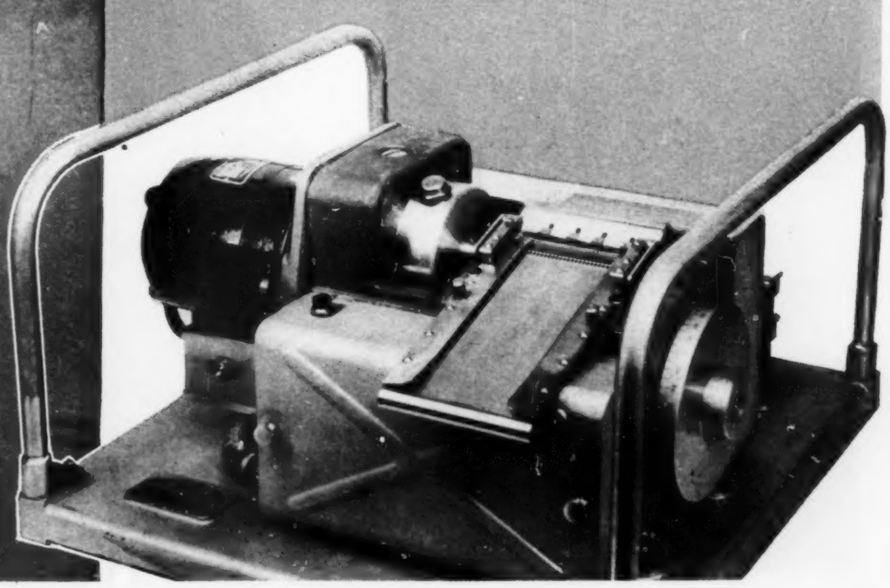
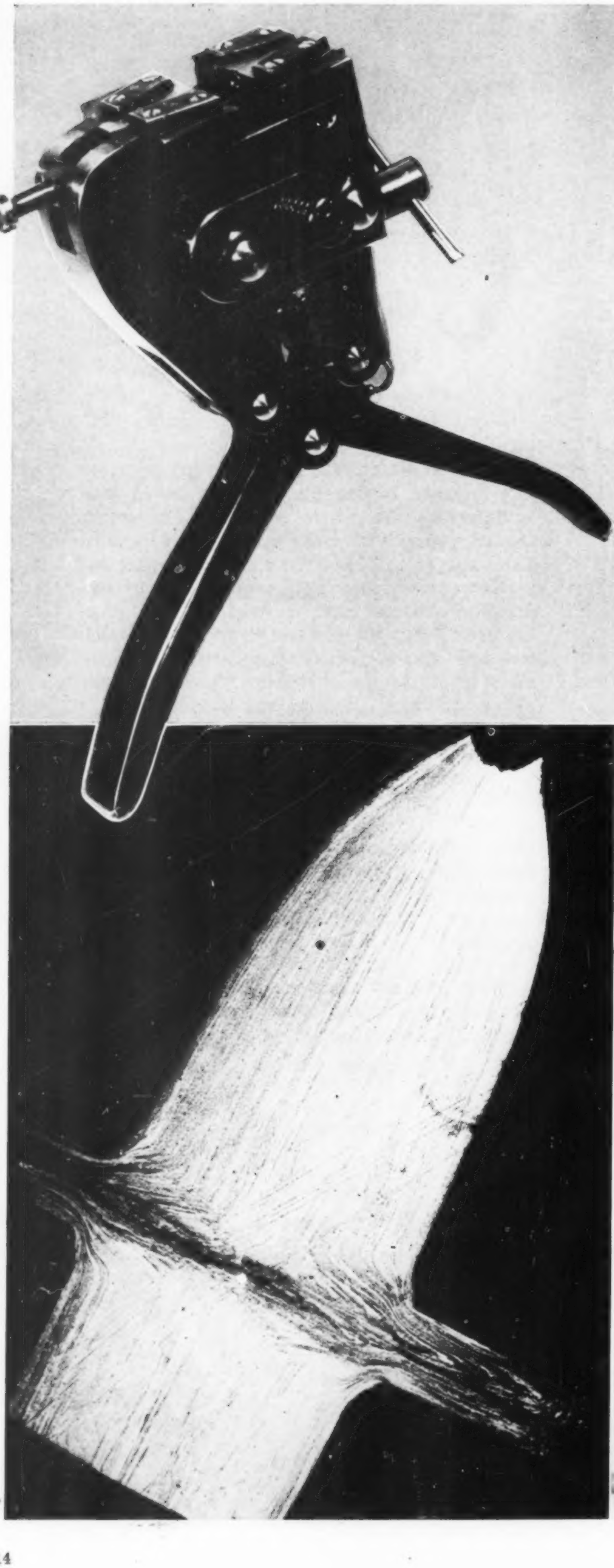




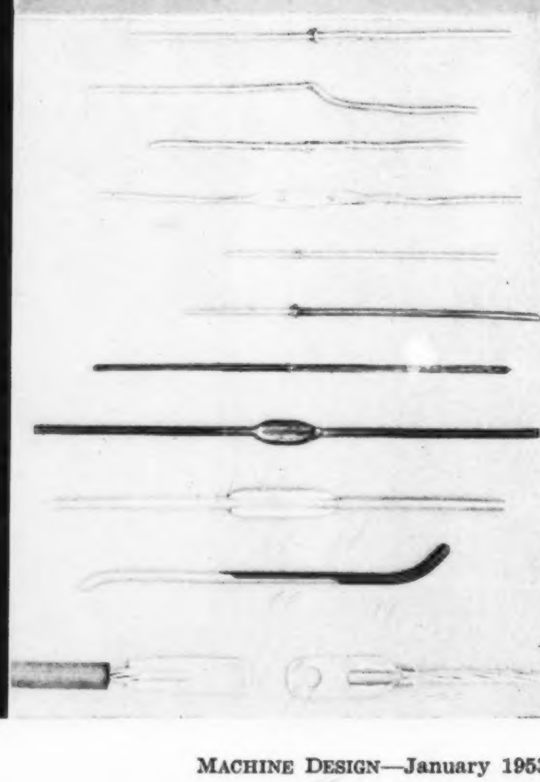
RECTANGULAR MOTION to produce accurate stepping movements is accomplished with an interesting device in the AMA Accounting Center reader, left. Designed by Bell Telephone Laboratories, this reader transforms information perforated on paper tapes into equivalent electrical information that may be properly processed. To step the perforated drum over which the tape rides sixteen times per second, the unusual step cam and U-shape follower shown above are used. In rotating, cam action is to first push the follower down to disengage the rack from the gear, then move the follower to the right and up into engagement, and finally to allow the step spring to pull the follower to the left for the desired rotational movement. To secure this cycle, the cam has two opposing 135-degree sectors of constant radius differing by an amount sufficient to permit the required accurate movement. Blending of the remaining 45-degree opposing sectors is done to produce virtually constant acceleration and deceleration. To prevent any possible movement except in the one-eighth cycle during stepping, a cam-activated lock arm holds the gear between steps.



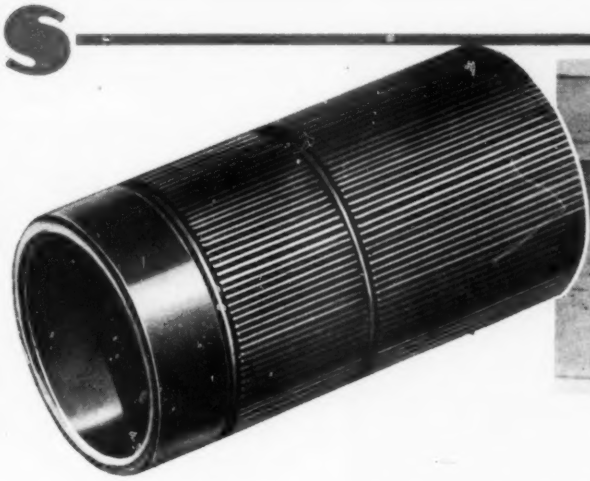


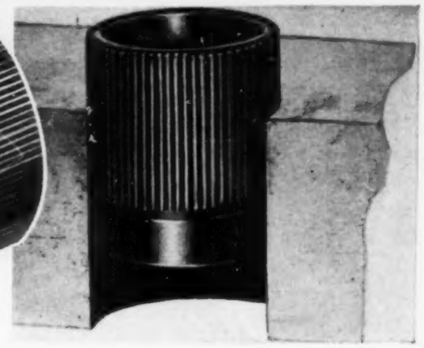


HEATLESS, FLUXLESS WELDING of nonferrous metals eliminates temperature effects and, unlike conventional welding, actually work-hardens the metal. Based on the fact that, when subjected to heavy pressure, metals will unite under intimate contact at the surfaces to form a homogeneous bond, this process has been developed primarily with those metals most amenable to molecular attraction—aluminum and copper. Under refinement by the Koldweld Corp., licensees to the General Electric Co. Ltd., England, this process shows promise of cost savings and improved joints for many applications. Some typical joints are shown below. Besides wire other shapes such as tubing, flat sheets, containers and other pieces, both light gage and heavy, have been welded as have other metals such as cadmium, lead, nickel, and zinc. In spite of the pressures necessary for successful welding, all but the largest sizes can be handled with toggle type hand tools, left, or kick press. Edge welds, roller seamed welds, ring seals, flanged tube joints, sheet to tube welds, wire mesh to sheet or mesh, and many other welds are possible.



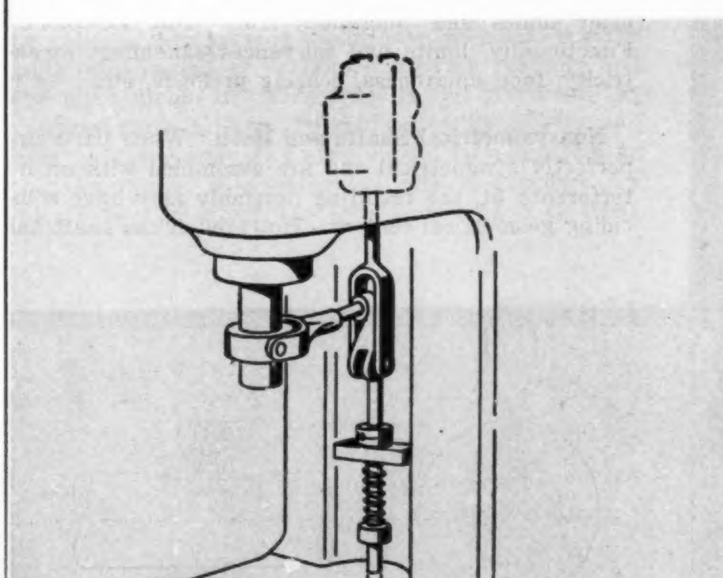
IDEAS





BROACH - FITTED BUSHINGS eliminate the need for close-tolerance limits on holes requiring secure locking. Made of hardened steel with a black oxide surface, these bushings have a broached OD which creates dozens of hard cutting edges. This permits forcing into castings, plastics, wrought steel, etc., without press fits. Hole tolerances as wide as 0.002-inch can be used. Manufactured by Aeroquip Corp., the bushings can be made with an extra broaching step where long fits are desirable or with a head flange for shouldering. Chip retainer grooves handle all excess material displaced in inserting.

TOUCH CONTROL over ram speed and force is obtained with a new servo type hydraulic valve on the Denison Multipress. As shown in the accompanying sketch, depression of the hand lever raises the shipper rod and, in turn, the valve spool through a clevis and yoke. Raising the valve spool opens the pressure port to move the ram down. Descent of the ram, however, pulls the telescoping arm attached to the yoke tending to return the valve spool to closed position until the hand lever is again moved. Close "feel" control with fast response is gained over the total ram stroke ranging from a minimum of 1/16-inch to a maximum of six to fifteen inches depending on press size.



HOLE FITS

Functional and economic factors in selection of tolerances and the resulting fits of shafts in holes

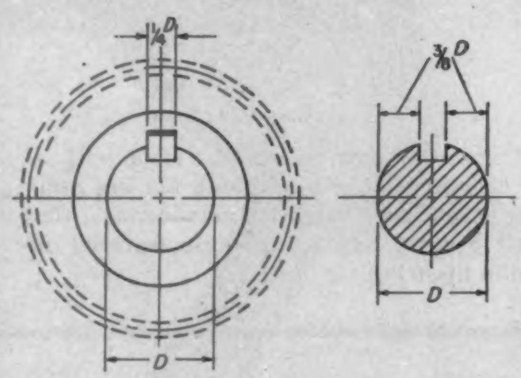


Fig. 1—Typical keyed shaft and hole arrangement with shaft showing unequal area of fit

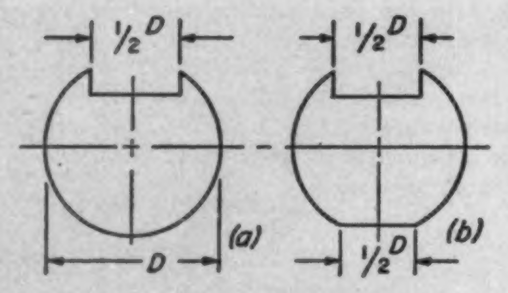


Fig. 2—Shaft with extrawide keyway, as at *a*, can be designed to have on opposite flat, as at *b*, to produce an accurate fit

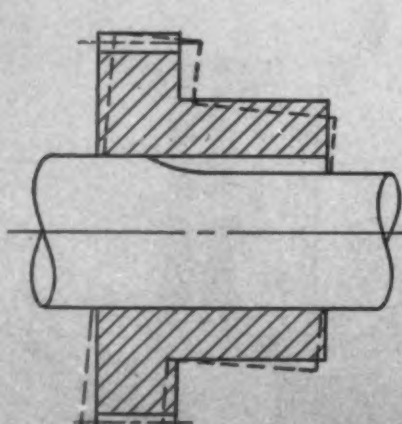


Fig. 3—Lack of symmetry in parts can produce a cocked fit on assembly as shown

GEOMETRICAL centers of holes and shafts do not always coincide when assembled, even though so shown on assembly drawings. From constantly seeing and drawing them in this manner a designer is apt to forget that only in a few cases does this condition exist.

An assembly drawing, as a rule, does not show the amount of clearance or interference that must be provided between holes and shafts. The determining of these values and of suitable tolerances is usually left to the man who makes the detail drawings. However, the success or failure mechanically or economically of a part often is determined by the intelligent choice of a dimensional tolerance of a few thousandths or even tenths of thousandths of an inch. Although mainly interested in functional aspects, the man who develops the design of the machine as a whole should also be interested in economy and, hence should carefully consider limits and tolerances from both viewpoints. Functionally, limits and tolerances can affect eccentricity, face squareness, bearing preloads, etc.

Nonsymmetrical Shafts and Holes: When parts are perfectly symmetrical and are assembled with an interference fit, the resulting assembly may have coinciding geometrical centers. However, if the shaft has

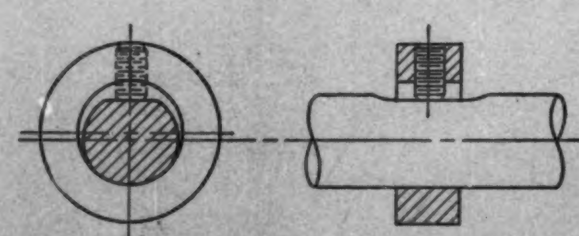


Fig. 4—Actual assembled condition produced with setscrew collar having oversize hole

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an appreciable flat or part of its area removed, it will crowd to one side when assembled. The same is true if the hole or shaft is provided with a keyway. Even an irregular hub shape can cause an eccentric assembly condition. Usually these amounts of eccentricity are too small to affect the functional characteristics of the assembly seriously but they should always be considered.

Average or standard width of a typical keyway is one-fourth the shaft diameter as shown in *Fig.* 1. Assuming dimensions of 2.0000 to 2.0008 inches for the hole and 2.0012 to 2.0020 inches for the shaft, and minimum hole size and the maximum shaft size, the metal interference would be two thousandths of an inch. With one-fourth the projected area missing on the keyway side, the shaft will offset itself eccentrically, when pressed in the hole, according to the ratios of the unbalanced projected areas, or 0.0003-inch. If the part is a precision gear of the shaved or ground tooth type, it would have an initial permissible total runout of 0.001-inch between the bore and the pitch diameter. This error is the maximum the designer expects in the finished assembly of gear and shaft. If, however, the gear should be assembled so that its runout is compounded with the eccentricity due to keyway effect, the total error would be 0.001 plus 0.0003 or 0.0013-inch. This is 30 per cent more than originally intended.

In the event the keyway is made abnormally wide because of requirements other than key load, *Fig. 2a*, this would cause an even more severe condition. To eliminate crowding of the shaft to one side, a relief of the same width can be placed opposite the keyway, as shown in *Fig. 2b*. Since an interference fit depends upon the elasticity of both parts and elasticity of metals is independent of hardness, this condition will exist regardless of the hardness of the metals involved.

Lack of symmetry of parts can cause an out-of-square condition as shown in *Fig.* 3. Here metal interference opposite the keyway is unbalanced by clearance at the keyway. The result is that the part, which should take a position square with the centerline of the shaft really assumes a cocked position shown exaggerated in *Fig.* 3. In actual practice the amount of tilt may be several thousandths and can be objectionable.

In all unbalanced interference fits, the greater the amount of metal interference, the greater the inaccuracy. Two remedies are available: (1) The design can be made symmetrical, which eliminates the cause of the distortion entirely, or; (2) the metal interference can be reduced, which does not eliminate the distortion entirely but does reduce it in direct proportion of the reduction of the metal interference.

When the shaft is smaller than the hole—a clearance fit—the geometrical centers of the hole and shaft will *never* coincide when assembled. If the clearance fit is a running fit, the rotation of the shaft will cause the lubricant to build up an oil wedge which will force the shaft to one side. With parts having clearance fits where there is no relative rotation between the shaft and hole, any clamping means such as a setscrew will create eccentricity. If the clamping method is not radial but longitudinal, as in the case of snap rings, any external radial force will move the bushing or other part into one-side contact with the shaft.

The average setscrew collar has a reamed hole square with the faces and is used on a shaft which is slightly smaller in diameter than the hole. The setscrew forces the collar to one side as shown in *Fig.* 4, causing the eccentric condition shown. Theoretically

Fig. 5—Light drive fit for parts to be disassembled may result in slight eccentricity

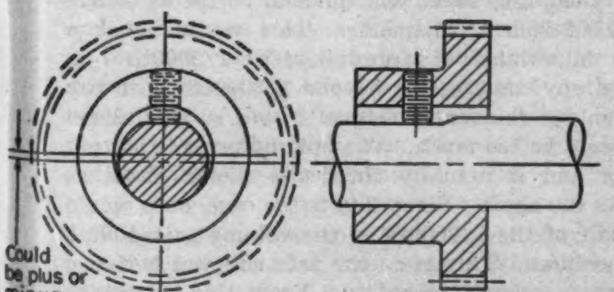
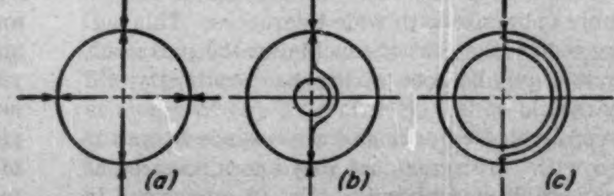


Fig. 6—Opposing forces of interference fit, *a*, are affected as at *b* by introduction of a small hole. Large hole *c* produces bushing characteristics with accompanying circumferential compression



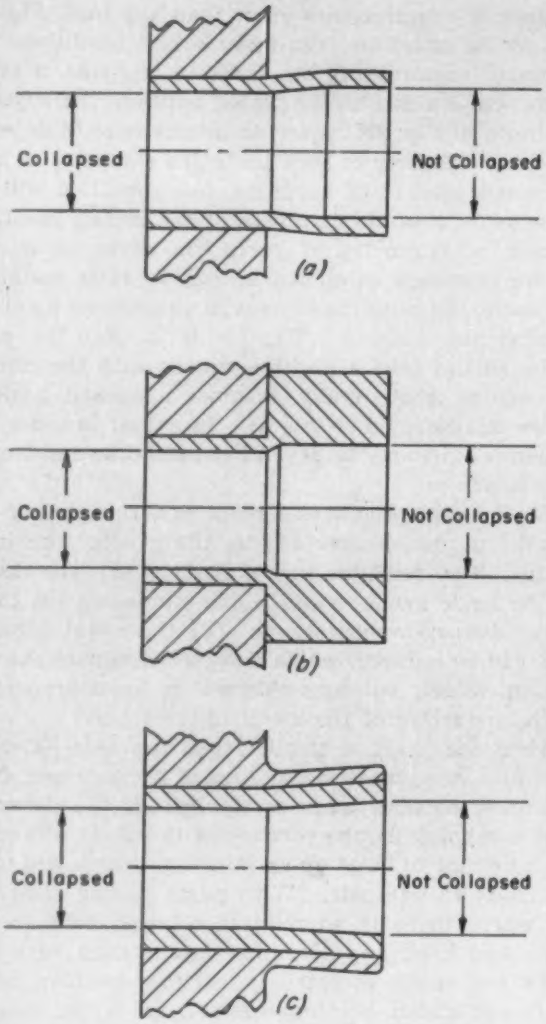


Fig. 7—Reduction of hole size during press fit of bushings can result in undesirable conditions, *a*, *b* and *c* wherein a portion of the bushing remains full size

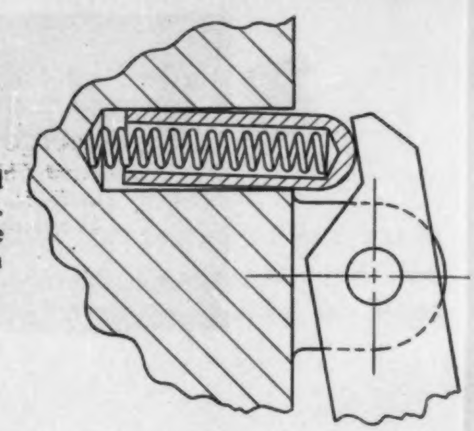


Fig. 8 — Spring and plunger arrangement. Wide-limit hole can be used if bore is smooth

only a line contact exists but actually, due to elasticity of parts, this is a small area and it is sufficient to square up the collar, if the face of the collar is square with the bore. Obviously, if the clearance is of the order of 4 or 5 thousandths instead of ½ to 1 thousandth, the same conditions are present and one fit is no better mechanically than the other. Consequently, collars can be specified to have low-class holes with wide tolerances and their mating shafts can be considerably undersize with wide tolerances. This will effectively reduce the cost of machining the parts but, of course, can only be done where the eccentricity will not be harmful.

If a gear or similar part has a clearance fit and is positioned with a setscrew, the clearances must be as small as possible to minimize the eccentricity. In cases where parts must be easily assembled and disassembled, it would be satisfactory to range from a small clearance to a metal-to-metal or snug fit. A very light drive fit would be preferable, however, if not objectionable from a shop standpoint. If a light drive fit is used there would still be a small amount of eccentricity due to the flat for the setscrew, *Fig.* 5.

Bushings and Liners: A combination of a clearance fit and an interference fit is used whenever a bushing or liner is pressed into a hole. A few of the fundamental effects of elasticity enter into this particular area and must be recognized. All metals are elastically compressible in volume up to their elastic limit. Soft steel, for instance, can be elastically compressed up to 1/3000 of its volume and, if stressed beyond this point remains permanently compressed. Therefore, any metal interference allowance greater than that required to develop an elastic press fit is wasted effort. Actually, to compress a steel shaft sufficiently to change its volume, the entire surface would have to be under pressure to eliminate escape sideways. The exact amount of this movement can be determined according to Poisson's ratio. In a hole and shaft of infinite length this side effect would be negligible or zero.

With these facts in mind a little calculation applied to a hypothetical case gives some interesting results. Suppose a soft-steel shaft of nominal 2-inch size is pressed into an unyielding hole 0.002-inch smaller, the unyielding hole being, of course, purely for the purpose of discussion. Since the hole by assumption cannot yield, the shaft will have to compress elastically 0.002-inch in diameter. This would cause a change in volume of approximately 1/2000th but, actually, any compression beyond 1/3000th of its volume would permanently deform it and so this allowance would be too much. The hole however is not unyielding and it actually elastically yields about as much as the shaft. Since this is the case, each would yield half of the 1/2000th of the volume or 1/4000th of the volume. This is on the safe side but with not too large a margin of safety. From this discussion it can readily be seen that any greater allowance for a press fit defeats its own purpose.

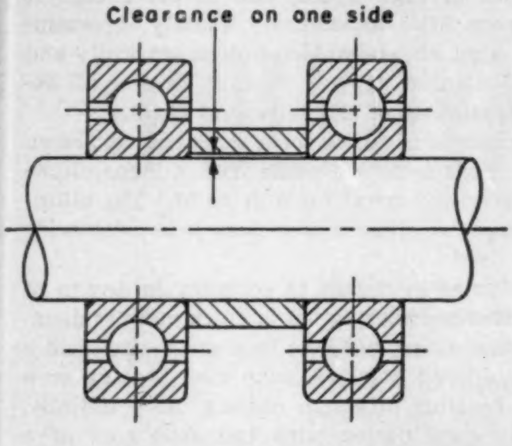


Fig. 9 — Left—Bearing spacing collar with adequate clearance can reduce costs owing to elimination of need for having the bore square with the parallel faces

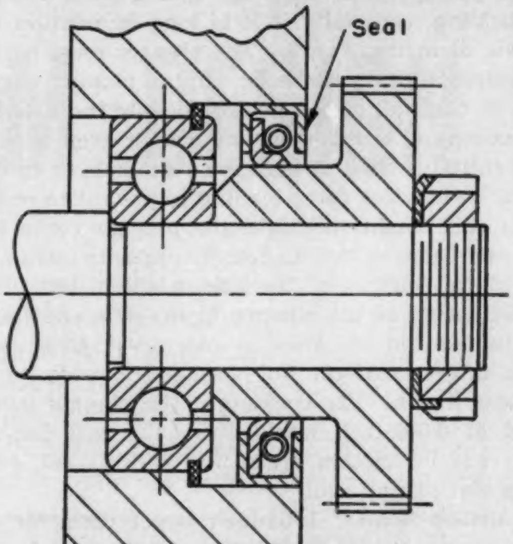
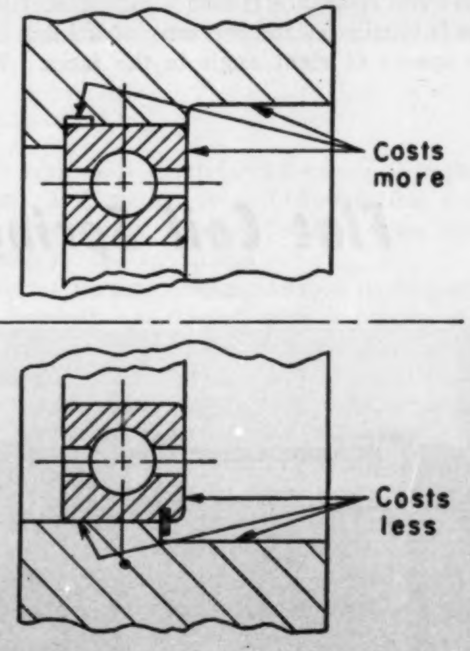


Fig. 10—Above—Low-class hole used with pressed-in seals can result in considerable savings if bearing type to allow a simple one-step hole is selected

Fig. 11 — Below — Comparison of bearing mounting hole designs with eye toward reducing production problems to a minimum



Opposing forces created by an interference fit are shown in *Fig. 6a.* Schematically they are shown in two directions only but are, of course, equally distributed in all directions. If a small hole is added, the condition shown in *Fig. 6b* arises. As the hole becomes larger it finally causes the forces to distribute themselves circumferentially around the bushing and, instead of elastically compressing radially as shown in *Fig. 6a,* they will compress circumferentially as shown in *Fig. 6c.* If the hole in the bushing is large in relation to its wall thickness these forces invariably collapse the bushing. The hole in the bushing is then reduced as much or sometimes more than the amount of the metal interference on the outside diameter of the bushing.

These facts should be considered when detailing bushings and shafts. The detail drawing of a bushing has the hole dimensioned to a standard that can be reamed and measured in production. Those dimensions will not exist after the bushing is pressed or driven in place. Therefore the shaft should have its dimensions based on what the dimensions will be after assembly, or the shop must remachine or reream the hole after assembly, to bring it back to standard drawing size. The choice is not optional but depends upon the shape of the parts involved and requires careful consideration.

In *Fig. 7a* part of the bushing is compressed and part is not. Obviously no shaft allowance will take care of such a condition. The bushing must be remachined after assembly by reaming or honing etc., and it should so state on the detail or assembly drawing or both. In *Fig 7b* the condition is similar to that shown at *a,* but the effect is exaggerated by the bushing having a thin wall and a large flange. At *c* is shown how a thin-wall boss surrounding a bushing can contribute to this condition.

Designing for Economy: The subject of economy in production is the most important consideration and it applies to *all* holes. It is axiomatic that if a drilled hole will do, instead of a wide-limit reamed hole, it should be used as it costs much less to produce. Likewise, a hole reamed to wide limits is cheaper to produce than one reamed to close limits. The relative economy of one hole over the other is still true even though the holes are finished by other means as grinding, boring, honing or lapping.

The function of the assembled parts should be carefully studied and the lowest cost hole that will allow the parts to function properly for the expected life of

the machine should be used. As an example a spring enclosed in a plunger can be used. The sole function of the plunger is to keep the spring from collapsing or buckling, especially if it is long in relation to its outside diameter, *Fig.* 8. The plunger must be a slip or clearance fit in the hole. If the plunger does not bear on the wall of the hole, obviously the hole is unnecessary and could be of any size or even be eliminated entirely. If it does bear on the hole as in *Fig.* 8, it will be due to a force tending to push it to one side. When this condition exists, the plunger cocks in the hole and bears at two diagonally opposite points. The rest of the surface of the hole is not in use. So far as the action of the plunger in the hole and its wear life in number of cycles is concerned, there will be little difference if the hole is made to wide limits or to close limits. The cocking of the plunger 0.009 instead of 0.001-inch would not make any discernible difference in friction, wear life, spring load, or anything else except cost.

A drilled hole of 1/64-inch larger diameter might even be used. It would have one drawback. Most drilled holes have appreciable ridges or tool marks in them. To make sure the sharp corner of the plunger could not catch on a ridge it would have to be chamfered. This additional operation on the plunger and its cost might more than offset the cost of rough reaming the hole to wide limits.

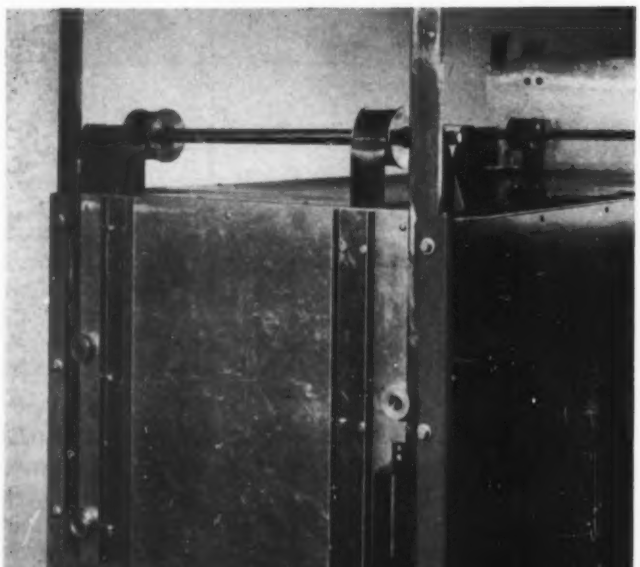
Another example is that of a spacing collar as shown in *Fig.* 9. The collar must have parallel faces and it must have a clearance as shown for ease of assembly. The clearance is also a manufacturing economy as it eliminates the necessity of holding the bore of the spacer at right angle to the faces. Whether that clearance is a small or a large amount will not affect any of the functions of an assembly of this kind except balance at high speed, and in percentage it would add very little unbalance. Usually in assemblies of this kind, the assembled unit is statically and dynamically balanced anyway, to take care of all accumulated unbalances of the individual parts.

Another example is a hole for a shaft seal as shown in *Fig.* 10. Most seals are made with a large allowance of interference metal for a drive fit. The allowance is enough to allow a low-class hole with wide limits to be used.

One more point in regard to economy in design of holes is worth remembering—it is much more economical to machine a through hole to a given size than a blind or shouldered hole. A snap ring bearing or a flange type bearing, although costing more initially, costs less in combination with the shop cost of a through hole, than a lower cost purchased part in combination with the shop cost of a blind or shouldered hole, *Fig.* 11. No tool can hold size close to a sharp corner and it is usually necessary to relieve the bottom of a shouldered hole in order that a bearing will seat properly against the face.

Although the subject of holes may appear to be of elementary scope, shop problems in hole production can be difficult, and serious thought given to proper specification can result in considerable advantage economically. The primary factor in design that must be kept in mind is that hole dimensions should never be copied promiscuously from tables or handbooks. Full consideration for the functioning of the particular parts being designed and suitable modification of fits for ease of production must be made.

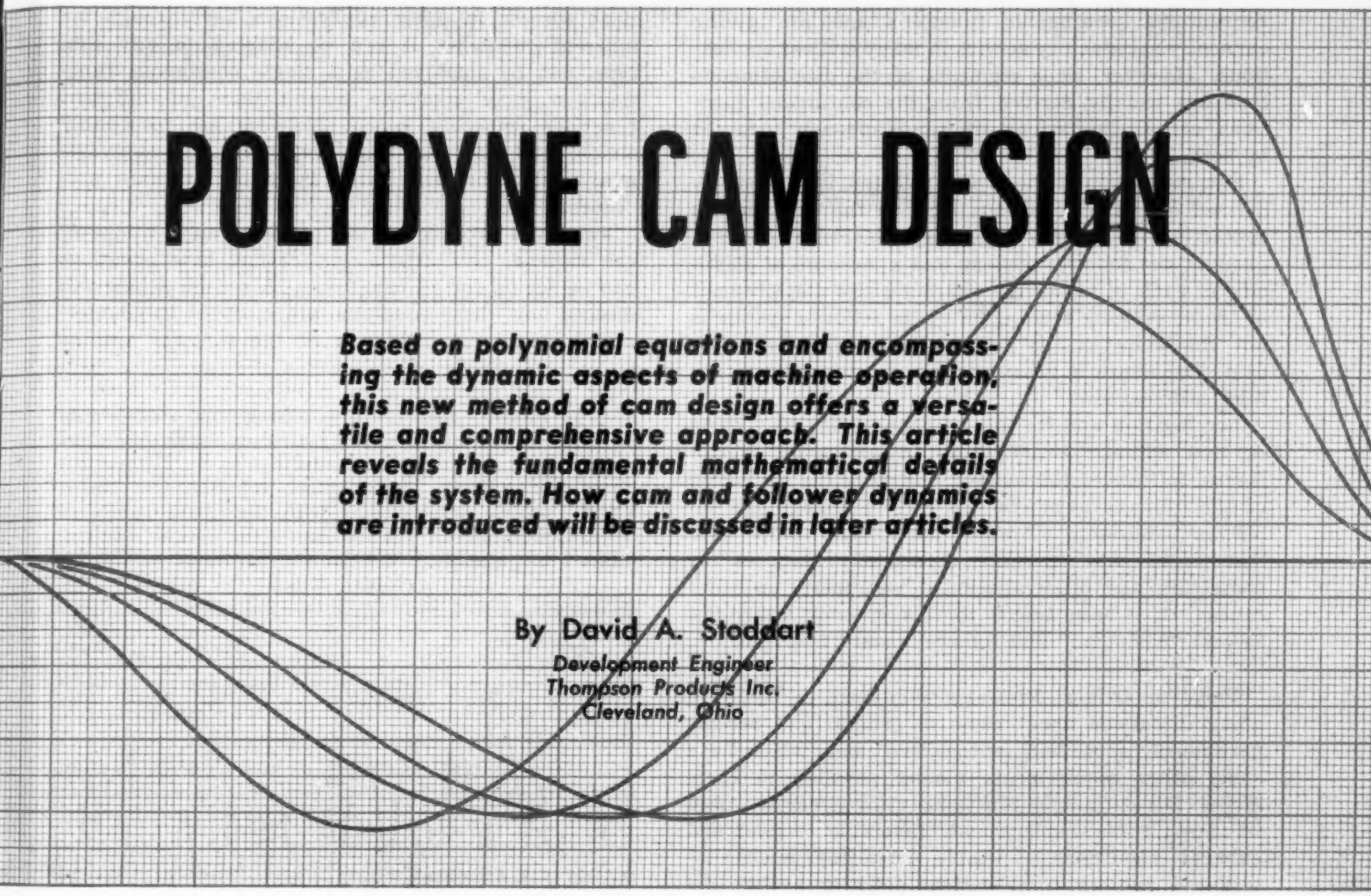
Flat Coil Springs Counterbalance Door



A PAIR of neg'ators are used to counterbalance the two 20-pound doors of the Magnaflux Zyglo drier. These constant-force spring elements, developed by the Hunter Spring Co., open and close the doors vertically and are mounted on bushings directly above the doors. Elimination of wire ropes, pulleys and countereweights has saved assembly time and reduced maintenance difficulties.

Strips of flat spring material which in their relaxed shape are tight coils, the 0.020 by 2.5-inch neg'ators are fastened to the inside of the door at the bottom; the remainder of the spring is stored on a 3-inch freely rotating bushing. They deliver a 20-pound force when extended 24 inches in closing the doors. Force is exerted as the highly stressed portion of the spring which is against the back of the door tends to return to its relaxed shape on the storage bushing. These neg'ators have been designed for a minimum of 10,000 operations at 300 F.

The drier is used to dry parts being inspected with Zyglo fluorescent penetrant, which is employed to detect critical defects



POLYDYNE CAM DESIGN

Based on polynomial equations and encompassing the dynamic aspects of machine operation, this new method of cam design offers a versatile and comprehensive approach. This article reveals the fundamental mathematical details of the system. How cam and follower dynamics are introduced will be discussed in later articles.

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A RATIONAL and flexible approach to cam design, the "polydyne" method embodies two fundamental features suggested by its coined title. They are the use of polynomial equations to define the cam contour and the accommodation in the contour of factors offsetting the dynamic characteristics of the cam and follower system.

Polydyne cam design has been evolved through successive stages over the last eight years in its application to automotive cams. Basic concepts proved successful in this field are potentially as well adaptable to all types of cams requiring control of kinematic properties.

Background: The continuing trend toward higher speeds has accented the need to eliminate or at least control the unfavorable phenomena of cam dynamics. Long recognized, acceleration of the follower system must be kept at a low value, since acceleration is a direct index of inertia forces ($F = ma$). This rule of nature undoubtedly led to formulation of the constant-acceleration profile and its ready acceptance. No other profile offers lower acceleration.

However, more than 20 years ago some designers observed that rate of change of acceleration must also be regarded with care. Instant application of acceleration, such as occurs with the constant-acceleration profile, or sudden changes in acceleration could be even more deleterious because of impact and vibration. Malfunction or self destruction were imminent as operating speed or follower mass increased under such profile conditions.

Additional factors causing serious deviations from results formerly anticipated with a simple idealized profile include slack or clearance, rigidity, and damping of the follower system. How some of these influences affect true cam action was demonstrated effectively several years ago by Hrones[1] who mathematically analyzed cam profiles of the type having a rise between two dwells. His study included the three conventional profiles—the constant-acceleration or parabolic, the simple-harmonic, and the so-called cycloidal, *Fig.* 1. Later confirmed experimentally by Mitchell,[2] results showed that vibratory deflections were far more severe for the constant-acceleration and the simple-harmonic profiles with their points of infinite rate of change of acceleration than for the cycloidal with its ever finite rate of change of acceleration.

In contrast to the cycloidal, which is classed as a continuous function, mixed or discontinuous functions[1, 3] have also been devised for eliminating infinite rate of change of acceleration. In one sense, discontinuous functions are most versatile for they permit far wider choice of profile design than the

[1] References are tabulated at end of article.

conventional continuous functions and are giving eminently satisfactory service in many critical applications. However, since segments of the discontinuous type are usually derived from the continuous functions, certain limitations are simultaneously incurred by both.

But, regardless of the profile selected, it has usually defined only the conditions local to the cam and its contacting follower. With any clearance between parts of the follower system, and a measure of elasticity in each of them, the characteristic displacement of the mass being acted upon by the follower system is not the same as the follower in contact with the cam. Departure is greatest when speed and mass are high, stiffness low.

Acceleration and its rate of change also play parts in this action. Acceleration, in producing inertia forces, influences the loading and thus the deflection of the follower members. Its rate of change can superpose additional vibratory deflections. Some phenomena of cam operation, previously subjects for conjecture and often attributed improperly to other causes, have been traced to this inherent difference in cam "command" and follower mass "response."

Polydyne Origin: Problems mentioned in the foregoing discussion about cams in general had become particularly acute with automotive cams, which have some other requirements almost peculiar to them alone. Discontinuous functions were generally used to obtain valve displacements favorable to inlet and exhaust conditions. Variable operating speed introduced another complication; follower system deflection varied with engine speed. Direct analytic approach in design was seldom successful. A final design was often reached only through trial and error.

In 1948, Dudley[4] first advanced several new concepts that gave promise of satisfying automotive cam requirements. He proposed first establishing the desired displacement of the valve, then introducing the deflection of the follower system, and finally determining the resulting contour of the cam.

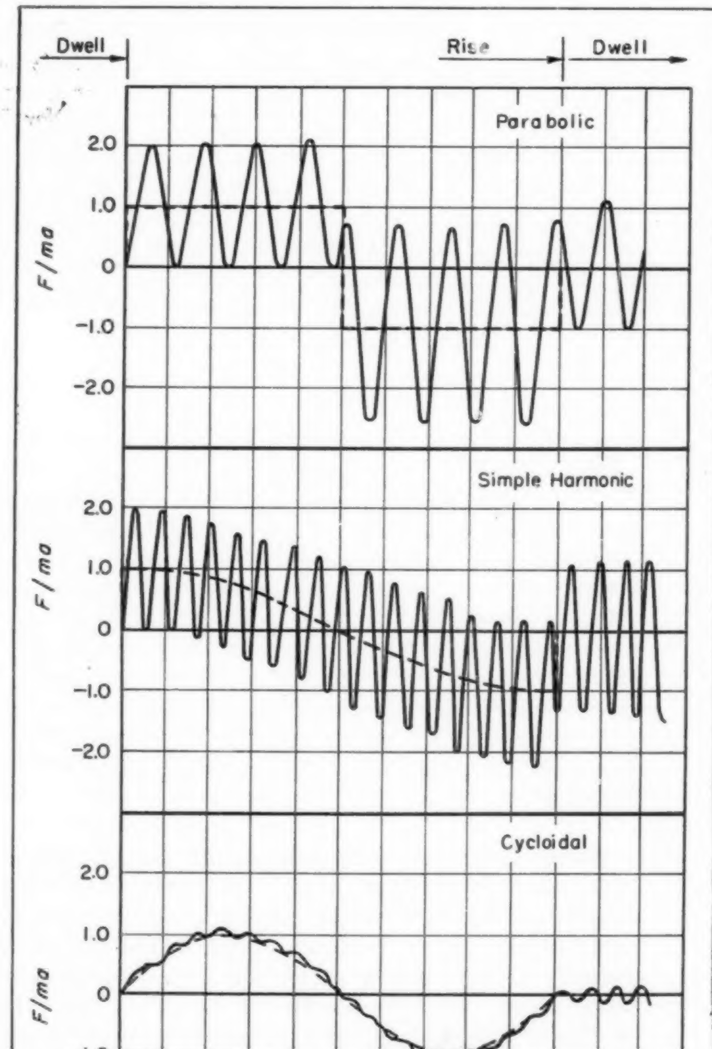
He turned to a versatile mathematical tool—polynomial equations—as the medium for expression of the required relationships. Briefly, the primary value of a polynomial equation is its limitless capacity to accommodate, through the addition of successive terms of different powers, as many conditions as are needed to define the action of the mechanism. Its other attributes in cam applications will be revealed later in more detail.

Based on Dudley's main proposals, work was continued at Thompson Products in evolving a successful polydyne method for automotive cam design. Essential features of the method were first announced two years ago[5] but details have not hitherto been published. Adequacy of the dual approach afforded by polydyne cam design has been confirmed by extensive testing. Moreover, cams designed by this method are being increasingly applied throughout the automotive industry.

Scope: Although polydyne design stands proved in only the automotive field to date, it should find equivalent success in other applications, for through its inherent flexibility it can be adapted to any cam type, can be employed as a single continuous function, or may comprise combinations of different functions smoothly joined. Methods are reasonably simple and the resulting properties can easily be analytically compared with those of alternate procedures.

Far from probing all the possible ramifications of the method, this article will outline how equations can be set up for all cam types. A second article will discuss how these equations are modified to counteract the expected deflection of the follower system.

Fig. 1—Ratio of force on cam arising from vibration to product of driven mass and maximum acceleration for no damping as determined analytically by Hrones for three cam contours. Dashed curves represent the "theoretical" profile accelerations with maximum values of ± 1. Comparatively the maximum theoretical accelerations are: parabolic, ± 4; simple harmonic ± 4.9; cycloidal, ± 6.3



A third and final article will develop an applied and fairly complex example.

Cam Types: Fundamentally, three types of cam actions are common. With their designations adopted for simplicity in later discussion, they are:

1. DRD—a rise both preceded and followed by dwells.
2. DRRD—a rise and immediate return, preceded and followed by dwells.
3. RRRR—a continuous sequence of rise, return, rise, etc.

These actions are schematically shown in ***Fig.*** **2.** Viewed comprehensively, each of these three types might be designed with either of two criteria in mind.

Minimum dynamic disturbance may be desired without regard for the displacement characteristics beween terminals of the event. An indexing cam typifies this common variation. The cam and follower must simply displace a mass from one point to another with minimum dynamic load.

Or, dynamic effects are allowed to rise above an absolute minimum to provide for an intended character of the displacement or some other property. This latter criterion is applied in automotive cams. Here, the cam's assignment is to lift the valve quickly to open the port and to return it quickly, so that maximum flow can occur in minimum time. Accordingly, acceleration is permitted to reach a far higher value than under the first criterion.

The three cam types, each with its two variations, are next discussed with respect only to the simple mathematics of the polynomial approach.

DRD Cams with Controlled Dynamics: The polynomial approach is most easily developed by starting with a primitive case, ***Fig.*** **3.** Incidentally, in this preliminary presentation of cam mathematics, a profile is considered for convenience to develop unity displacement in unity time, and will be portrayed in Cartesian co-ordinates. Again for mathematical convenience, the initial rise will be shown at the right so that a second cam phase, or return, can be conveniently depicted to the left in the second quadrant.

Assume that the purpose of the profile in ***Fig.*** **3** is simply to displace the follower from $y = 0$ to $y = 1$ as the point of contact moves from $x = 1$ to $x = 0$. An elementary polynomial equation defines the displacement:

$$y = C_0 + C_1x$$

By inspection, the coefficients are easily evaluated and

$$y = 1 - x$$

It is just as easy to see that this cam is also kinematically primitive. As ***Fig.*** **3** shows, at the event terminals finite velocity changes are instantaneous and acceleration is infinity. This cam, as poor as it is, neverthless is the lowly antecedent of all polynomial profiles.

The next step in the evolution of a kinematically favorable profile is to control the terminal velocity as well as the displacement. Then, ***Fig.*** **4,** the assumed conditions are: When $x = 1$; $y = 0$, $y' = 0$. When $x = 0$; $y = 1$, $y' = 0$. Throughout this discussion the prime notation will be employed to designate successive derivatives of displacement y. That is, $y' = dy/dx$ = velocity; $y'' = d^2y/dx^2$ = acceleration; etc. Since for the example of ***Fig.*** **4** there are now four conditions, the polynomial must be set up with four unknown coefficients:

$$y = C_0 + C_1x + C_2x^2 + C_3x^3$$

Also, since velocity conditions have been introduced, the velocity equation—the first derivative of y—must also be written:

$$y' = 0 + C_1 + 2C_2x + 3C_3x^2$$

Inserting the four assumed conditions in these two general equations gives:

$$\begin{array}{ll}
(x = 1, y = 0) & 0 = C_0 + C_1 + C_2 + C_3 \\
(x = 0, y = 1) & 1 = C_0 \\
(x = 1, y' = 0) & 0 = C_1 + 2C_2 + 3C_3 \\
(x = 0, y' = 0) & 0 = C_1
\end{array}$$

With two coefficients evaluated, the first and third equations are

$$0 = 1 + C_2 + C_3$$
$$0 = 2C_2 + 3C_3$$

Simultaneous solution of these equations gives $C_2 = -3$, $C_3 = 2$. The specific displacement, velocity and acceleration equations are shown in ***Fig.*** **4.** These

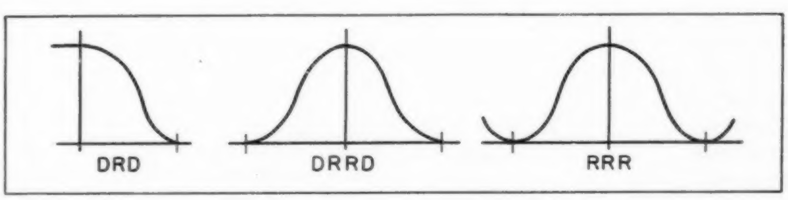


Fig. 2—Representations and designations for three basic cam types: dwell - rise - dwell, dwell - rise - return - dwell, and rise - return - rise

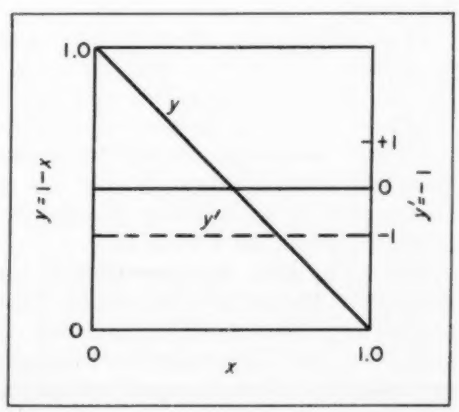


Fig. 3—Right — Primitive antecedent of polynomial cam contours, resulting when only displacement is controlled. Acceleration is infinity at both ends of the event

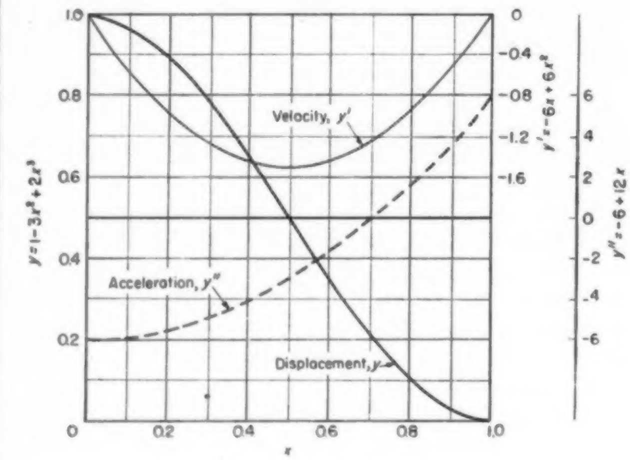


Fig. 4—Second evolutionary stage of satisfactory polynomial contours. Both displacement and velocity are controlled. Acceleration is finite, but its rate of change, jerk, is infinite at both event terminals

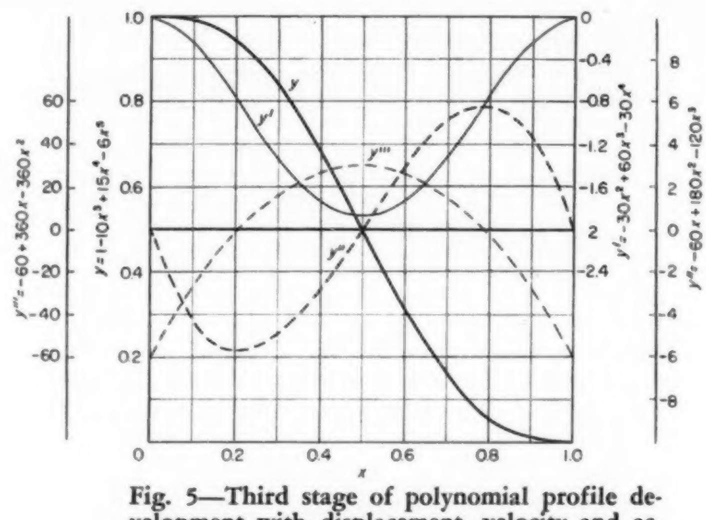


Fig. 5—Third stage of polynomial profile development with displacement, velocity and acceleration all controlled. Both acceleration and its rate of change, jerk, are finite over the entire event

equations reveal that acceleration is limited to a finite range. However, it has the same fault as the conventional constant-acceleration and simple-harmonic profiles. Acceleration is instantly applied at the terminals of the event, producing shock loading.

The obvious extension of this procedure is to impose additional conditions restricting acceleration to zero at the event terminals. Accordingly when $x = 1$; $y = 0$, $y' = 0$, $y'' = 0$. When $x = 0$; $y = 1$, $y' = 0$, $y'' = 0$. And, the general displacement polynomial, to accommodate six conditions, becomes:

$$y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5$$

The general velocity and acceleration equations, also required for determination of the coefficients, are by simple differentiation:

$$y' = C_1 + 2\,C_2x + 3\,C_3x^2 + 4\,C_4x^3 + 5\,C_5x^4$$
$$y'' = 2\,C_2 + 6\,C_3x + 12\,C_4x^2 + 20\,C_5x^3$$

Substituting the assumed six conditions in these equations permits simultaneous solution for the coefficients. The final displacement equation and curve are shown in *Fig.* 5 with plots of the first three derivatives. This profile has previously been suggested by Gutman.[6]

This polynomial offers rather favorable dynamic properties; it compares well with the cycloidal form, *Fig.* 1. Without a searching analysis they can almost safely be accepted as equivalent forms. Offsetting the slight advantage of the 5.8 acceleration for the 3-4-5 polynomial, versus 6.3 for the cycloidal, is the higher rate of acceleration change, or "jerk," at the event terminals, 60 versus 40.

The term "jerk" will hereafter be used to designate the rate of change of acceleration. Because of its repeated use, convenient identification of this property is helpful, but unfortunately academicians have not yet assigned a formal, equally descriptive label.

Table 1—Coefficient Evaluations

(for case 2-C: control at $x = 1$, and displacement derivatives = 0)

Polynomial: $y = C_0 + C_px^p + C_qx^q + C_rx^r + C_sx^s + \cdots$

$$C_p = -\frac{C_0\,qrs\,\cdots}{(q-p)(r-p)(s-p)\,\cdots}$$

$$C_q = -\frac{C_0\,prs\,\cdots}{(p-q)(r-q)(s-q)\,\cdots}$$

$$C_r = -\frac{C_0\,pqs\,\cdots}{(p-r)(q-r)(s-r)\,\cdots}$$

$$C_s = -\frac{C_0\,pqr\,\cdots}{(p-s)(q-s)(r-s)\,\cdots}$$

Example: $y = 1 + C_4x^4 + C_5x^5 + C_6x^6 + C_7x^7$

$$C_4 = -\frac{(1)(5)(6)(7)}{(1)(2)(3)} = -35$$

$$C_5 = -\frac{(1)(4)(6)(7)}{(-1)(1)(2)} = 84$$

$$C_6 = -\frac{(1)(4)(5)(7)}{(-2)(-1)(1)} = -70$$

$$C_7 = -\frac{(1)(4)(5)(6)}{(-3)(-2)(-1)} = 20$$

Solution: $y = 1 - 35x^4 + 84x^5 - 70x^6 + 20x^7$

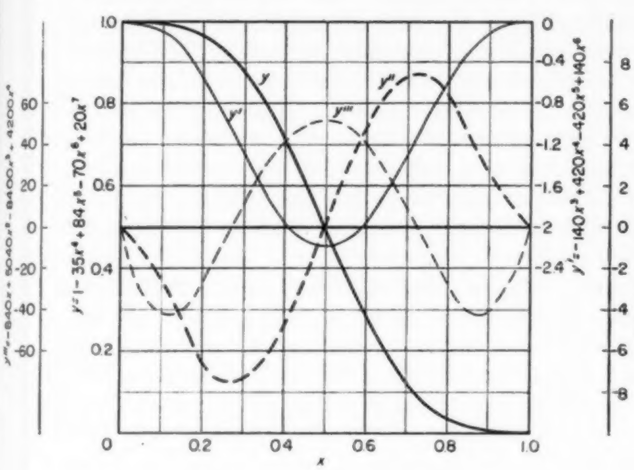


Fig. 6—A polynomial profile with zero jerk at the event terminals. Maximum acceleration is somewhat greater than for lower-order forms and is actually tangential to the x-axis

Discussion of differences between the 3-4-5 polynomial and the cycloidal is probably only speculation, for they are so subtle that they mean little in terms of actual cams and cam producing methods. However, the comparison is being stressed for a practical reason of another sort. When the "dyne" part of the design procedure is later described, ease of incorporating system characteristics into a polynomial form, in contrast to any other, will be clearly evident. The "dyne" phase is not necessarily restricted to polynomials, but its application to other types of functions is sometimes not possible by rational mathematical methods and at best, more difficult.

The point of diminishing returns is being approached in this sequence of successive derivative control, but perhaps it has not yet been reached. Controlling jerk at both terminals of the event is the next step. That is, when $x = 1$; $y = 0$, $y' = 0$, $y'' = 0$, $y''' = 0$. When $x = 0$; $y = 1$, $y' = 0$, $y'' = 0$, $y''' = 0$. These eight conditions establish the general polynomial:

$$y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5 + C_6x^6 + C_7x^7$$

To reduce the formidable appearance of larger polynomials such as these, certain coefficients might be immediately evaluated—those that equal zero. Such coefficients are easily spotted. If, when $x = 0$, $d^ny/dx^n = 0$, then $C_n = 0$. This truth is demonstrated for polynomials so far developed by the following tabulation which is extended to include the seventh-power equation just proposed.

Initial Powers	Zero Derivative (at $x=0$)	Final Powers (with finite coeff.)
0 and 1	....	0-1
0 to 3	y'	0-2-3
0 to 5	y', y''	0-3-4-5
0 to 7	y', y'', y'''	0-4-5-6-7

Also, C_0 is usually given by the assigned conditions, since it is equal to the lift—the total displacement. That is, the condition—when $x = 0$, $y = 1$ — means $C_0 = 1$. Therefore, instead of the foregoing equation with its eight coefficients, a simpler polynomial can be written:

$$y = 1 + C_4x^4 + C_5x^5 + C_6x^6 + C_7x^7$$

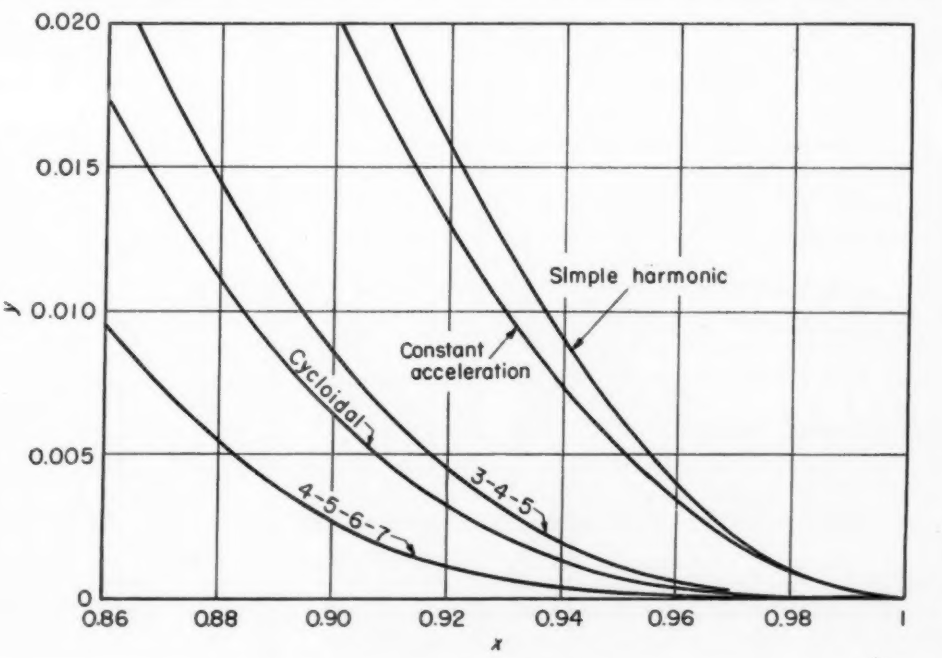


Fig. 7—Comparison of displacement characteristics at the initial end of the event for conventional and several polynomial profiles. Graph range includes only 2 per cent of the lift, 14 per cent of the cam rotation required for the event

With the remaining assigned conditions, coefficients C_4 to C_7 can be found by simultaneous solution as before. However, as the polynomial terms increase beyond two or three, simultaneous solution becomes more tedious and more susceptible to error. This problem is eliminated by simplified formulation of coefficients shown in TABLE 1 which includes example solutions for the coefficients of the 4-5-6-7 polynomial.

Finally, the resulting equations and pertinent curves for the 4-5-6-7 polynomial can be developed, *Fig.* 6. Besides the curves of *Fig.* 6, the following data help evaluate the significance of the 4-5-6-7 polynomial:

Profile	Max. Accel. y''	Max. Jerk y'''	Max. Jerk Deriv. y^{iv}
Const. Acc.	± 4	$\pm\infty$	......
Harmonic	± 4.9	$\pm\infty$	......
3-4-5	± 5.8	$+30, -60$	∞
Cycloidal	± 6.3	± 40	∞
4-5-6-7	± 7.3	$+52, -42$	840

The physical meaning of these figures is best depicted by a large-scale plotting of the five profiles, *Fig.* 7. Besides the slower rate of application of acceleration at the event terminals, and thus of inertia forces, an advantage of such a profile as the 4-5-6-7 in some applications is the more gradual beginning lift. Hence, where timing of related functions is critical, less sensitive set up or adjustment is required relative to follower displacement. Vibratory properties of this form have not been investigated, but they are quite likely not much inferior to the cycloidal or the 3-4-5 polynomial, although dynamic loading is obviously slightly greater.

In similar fashion, the fourth and higher derivatives of displacement can be specified. As this process continues, however, nature exacts a toll: maximum acceleration continues to rise. The technique is possibly appropriate only in certain cases to be treated later.

DRD Cams with Specified Displacements: In the previous section, polynomials with successive integral power terms were constructed to show a method for obtaining a measure of control over acceleration and

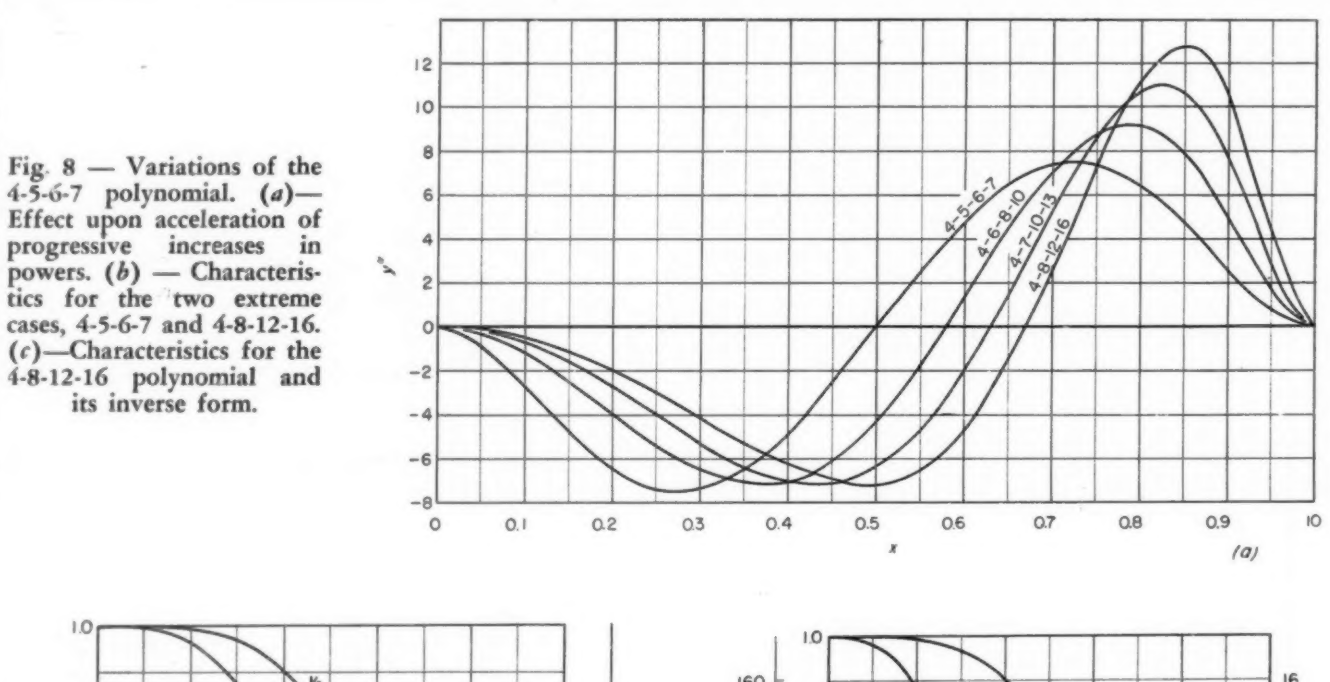


Fig. 8 — Variations of the 4-5-6-7 polynomial. (*a*)—Effect upon acceleration of progressive increases in powers. (*b*) — Characteristics for the two extreme cases, 4-5-6-7 and 4-8-12-16. (*c*)—Characteristics for the 4-8-12-16 polynomial and its inverse form.

its rate of change. The process can be easily varied to obtain a desired characteristic of velocity or displacement. The latter property is perhaps the one most usually of concern and is taken as the object of discussion here. A similar technique would be effective for varying the velocity characteristics.

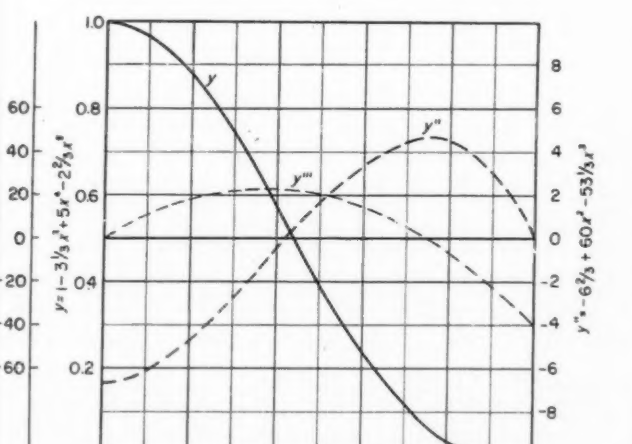
Actually, there is no convenient method for specifying the required displacement curve and then synthesizing it by the polynomial approach. Perhaps on the surface there appears to be a rational procedure involving the assignment of co-ordinates to a number of points along the displacement curve and the establishment of a polynomial to satisfy these conditions as well as those that must perforce still be assigned at the terminals of the event. However, mathematics of such a process quickl ybecome overwhelming. Frustration is avoided by the simple method of plotting a number of displacement curves and their associated acceleration curves, and progressively varying the powers until the desired curve is approached with sufficient accuracy. If the desired curve cannot be closely approximated by this method, the chances are that its dynamic properties are not only uncontrolled but intolerable.

Different displacement curves can be obtained by using the polynomial method already outlined except that different powers are employed. An example quickly describes the technique.

Assume that the same initial conditions set up for the 4-5-6-7 polynomial having controlled dynamic properties are again employed. That is, when $x = 1$; $y = 0$, $y' = 0$, $y'' = 0$, $y''' = 0$. When $x = 0$; $y = 1$, $y' = 0$, $y'' = 0$, $y''' = 0$. Then, with the known simplification of the 4-5-6-7 polynomial expressed in generalized form,

$$y = C_0 + C_p x^p + C_q x^q + C_r x^r + C_s x^s$$

Fig. 9—Simplest polynomial for DRRD profiles having zero acceleration at the start of the event and zero velocity with finite acceleration at juncture of rise and return phases



But, let p, q, r, and s have different sets of values, such as:

p	q	r	s
4	5	6	7
4	6	8	10
4	7	10	13
4	8	12	16

Evaluation of the coefficients proceeds exactly as before with the aid of TABLE 1. Resulting displacement, acceleration, and jerk curves are plotted in *Fig.* 8. The effect of higher powers is easily observed. The higher the powers, the more area under the displacement curve, and the greater the maximum acceleration and jerk which both shift toward the start of the rise.

Curves of the same powers can be built up on the other side of the 4-5-6-7 curve by the simple expedient of inverting the calculated points. That is, $x_i = 1-x$ and $y_i = 1-y$ where x, y are the calculated co-ordinates and x_i, y_i are the inverted co-ordinates. Such inversion is shown in dashed lines for the 4-8-12-16 curve. For velocity y' only x values would be inverted. Acceleration and jerk curves for the inverted displacement curve are also shown dashed in *Fig.* 8.

The process of power selection can continue ad infinitum, of course, and there need be no fixed increment between the powers. However, application of powers according to some systematic plan usually facilitates interpolation toward the powers which will most closely yield the desired displacement curve. Just as the foregoing example demonstrates variations of the 4-5-6-7 polynomial, so can similar methods be used with the 3-4-5 or any basic polynomial of more terms.

Laboriousness of calculations can be reduced by several means. Work sheet methods can be adopted as later demonstrated. A table of powers is almost a necessity. One containing most powers of practical use in cam design for numbers from 0.10 to 0.99 is included in this article, TABLE 2. Powers of factors from 0.01 to 0.09 can be obtained, of course, simply by moving the decimal point.

DRRD Cams: Building polynomial equations for cam systems having dwell, rise, immediate return, and dwell is the same as for their DRD counterparts except for the assigned conditions.

Kinematics dictate again that displacement, velocity, and acceleration must be zero at the event terminals, but at the midevent point, velocity should be zero, displacement, of course, is 1, and acceleration should be unspecified. Or, when $x = 1$; $y = 0$, $y' = 0$, $y'' = 0$. When $x = 0$; $y = 1$, $y' = 0$. It will be assumed that the cam is symmetrical and therefore that the first half-event characteristics (first quadrant) will be duplicated with x inverted in the second half event (second quadrant).

Additionally, if acceleration is to be smoothly continuous across the axis dividing the first and second quadrants, its rate of change must be zero. Therefore, another condition is that when $x = 0$, $y''' = 0$.

Table 2—Powers 2 to 30

No.	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
10	0100	00 1000	0001 0000	0000 1000	0000 0100	0000 0010	0000 0001								
11	121	1331	1 4641	1610	177	19	2								
12	144	1728	2 0736	2488	298	35	4								
13	169	2197	2 8561	3712	482	62	8	1							
14	196	2744	3 8416	5378	752	105	14	2							
15	225	3375	5 0625	7593	1139	170	25	3							
16	256	4096	6 5536	1 0485	1677	268	42	6	1						
17	289	4913	8 3521	1 4198	2413	410	69	11	2						
18	324	5832	10 4976	1 8895	3401	612	110	19	3						
19	361	6859	13 0321	2 4760	4704	893	169	32	6	1					
20	0400	00 8000	0016 0000	0003 2000	0000 6400	0000 1280	0000 0256	0000 0051	0000 0010	0000 0002					
21	441	9261	19 4481	4 0841	8576	1801	378	79	16	3					
22	484	1 0648	23 4256	5 1536	1 1337	2494	548	120	26	5	1				
23	529	1 2167	27 9841	6 4363	1 4803	3404	783	180	41	9	2				
24	576	1 3824	33 1776	7 9626	1 9110	4586	1100	264	63	15	3				
25	625	1 5625	39 0625	9 7656	2 4414	6103	1525	381	95	23	5	1			
26	676	1 7576	45 6976	11 8813	3 0891	8031	2088	542	141	36	9	2			
27	729	1 9683	53 1441	14 3489	3 8742	1 0460	2824	762	205	55	15	4	1		
28	784	2 1952	61 4656	17 2103	4 8189	1 3492	3778	1057	296	82	23	6	1		
29	841	2 4389	70 7281	20 5111	5 9482	1 7249	5002	1450	420	122	35	10	2		
30	0900	02 7000	0081 0000	0024 3000	0007 2900	0002 1870	0000 6561	0000 1968	0000 0590	0000 0177	0000 0053	0000 0015	0000 0004	0000 0001	
31	961	2 9791	92 3521	28 6291	8 8750	2 7512	8528	2643	819	254	78	24	7	2	
32	1024	3 2768	104 8576	33 5544	10 7374	3 4359	1 0995	3518	1125	360	115	36	11	3	1
33	1089	3 5937	118 5921	39 1353	12 9146	4 2618	1 4064	4641	1531	505	166	55	18	5	1
34	1156	3 9304	133 6336	45 4354	15 4480	5 2523	1 7857	6071	2064	701	238	81	27	9	3
35	1225	4 2875	150 0625	52 5218	18 3826	6 4339	2 2518	7881	2758	965	337	118	41	14	5
36	1296	4 6656	167 9616	60 4661	21 7678	7 8364	2 8211	1 0155	3656	1316	473	170	61	22	7
37	1369	5 0653	187 4161	69 3439	25 6572	9 4931	3 5124	1 2996	4808	1779	658	243	90	33	12
38	1444	5 4872	208 5136	79 2351	30 1093	11 4415	4 3477	1 6521	6278	2385	906	344	130	49	18
39	1521	5 9319	231 3441	90 2241	35 1874	13 7231	5 3520	2 0872	8140	3174	1238	482	38	73	28
40	1600	06 4000	0256 0000	0102 4000	0040 9600	0016 3840	0006 5536	0002 6214	0001 0485	0000 4194	0000 1677	0000 0671	0000 0268	0000 0107	0000 0042
41	1681	6 8921	282 5761	115 8562	47 5010	19 4754	7 9849	3 2738	1 3422	5503	2256	925	379	155	63
42	1764	7 4088	311 1696	130 6912	54 8903	23 0539	9 6826	4 0667	1 7080	7173	3012	1265	531	223	93
43	1849	7 9507	341 8801	147 0084	63 2136	27 1818	11 6882	5 0259	2 1611	9292	3995	1718	738	317	136
44	1936	8 5184	374 8096	164 9162	72 5631	31 9277	14 0482	6 1812	2 7197	1 1966	5265	2316	1019	448	197
45	2025	9 1125	410 0625	184 5281	83 0376	37 3669	16 8151	7 5668	3 4050	1 5322	6895	3102	1396	628	282
46	2116	9 7336	447 7456	205 9629	94 7429	43 5817	20 0476	9 2219	4 2420	1 9513	8976	4129	1899	873	401
47	2209	10 3823	487 9681	229 3450	107 7921	50 6623	23 8112	11 1913	5 2599	2 4721	1 1619	5460	2566	1206	566
48	2304	11 0592	530 8416	254 8039	122 3059	58 7068	28 1792	13 5260	6 4925	3 1164	1 4958	7180	3446	1654	794
49	2401	11 7649	576 4801	282 4752	138 4128	67 8223	33 2329	16 2841	7 9792	3 9098	1 9158	9387	4599	2253	1104
50	2500	12 5000	0625 0000	0312 5000	0156 2500	0078 1250	0039 0625	0019 5312	0009 7656	0004 8828	0002 4414	0001 2207	0000 6103	0000 3051	0000 1525
51	2601	13 2651	676 5201	345 0252	175 9628	89 7410	45 7679	23 3416	11 9042	6 0711	3 0962	1 5791	8053	4107	2094
52	2704	14 0608	731 1616	380 2040	197 7060	102 8071	53 4597	27 7990	14 4555	7 5168	3 9087	2 0325	1 0569	5496	2857
53	2809	14 8877	789 0481	418 1954	221 6436	117 4711	62 2596	32 9976	17 4887	9 2690	4 9125	2 6036	1 3799	7313	3876
54	2916	15 7464	850 3056	459 1650	247 9491	133 8925	72 3019	39 0430	21 0832	11 3849	6 1478	3 3198	1 7927	9680	5227
55	3025	16 6375	915 0625	503 2843	276 8064	152 2435	83 7339	46 0536	25 3295	13 9312	7 6621	4 2141	2 3178	1 2747	7011
56	3136	17 5616	983 4496	550 7317	308 4097	172 7094	96 7173	54 1616	30 3305	16 9851	9 5116	5 3265	2 9828	1 6703	9354
57	3249	18 5193	1055 6001	601 6920	342 9644	195 4897	111 4291	63 5146	36 2033	20 6358	11 7624	6 7046	3 8216	2 1783	1 2416
58	3364	19 5112	1131 6496	656 3567	380 6869	220 7984	128 0630	74 2765	43 0804	24 9866	14 4922	8 4055	4 8751	2 8276	1 6400
59	3481	20 5379	1211 7361	714 9242	421 8053	248 8651	146 8304	86 6299	51 1116	30 1558	17 7919	10 4972	6 1933	3 6540	2 1550
60	3600	21 6000	1296 0000	0777 6000	0466 5600	0279 9360	0167 9616	0100 7769	0060 4661	0036 2797	0021 7678	0013 0606	0007 8364	0004 7018	0002 8211
61	3721	22 6981	1384 5841	844 5963	515 2037	314 2742	191 7073	116 9414	71 3342	43 5139	26 5434	16 1915	9 8768	6 0248	3 6751
62	3844	23 8328	1477 6336	916 1328	568 0023	352 1614	218 3401	135 3708	83 9299	52 0365	32 2626	20 0028	12 4017	7 6890	4 7672
63	3969	25 0047	1575 2961	992 4365	625 2350	393 8980	248 1557	156 3381	98 4930	62 0506	39 0918	24 6278	15 5155	9 7748	6 1581
64	4096	26 2144	1677 7216	1073 7418	687 1947	439 8046	281 4749	180 1439	115 2921	73 7869	47 2236	30 2231	19 3428	12 3794	7 9228
65	4225	27 4625	1785 0625	1160 2906	754 1889	490 2227	318 6448	207 1191	134 6274	87 5078	56 8800	36 9720	24 0318	15 6206	10 1534
66	4356	28 7496	1897 4736	1252 3325	826 5395	545 5160	360 0406	237 6268	156 8336	103 5102	68 3167	45 0890	29 7587	19 6407	12 9629
67	4489	30 0763	2015 1121	1350 1251	904 5838	606 0711	406 0676	272 0653	182 2837	122 1301	81 8271	54 8242	36 7322	24 6105	16 4890
68	4624	31 4432	2138 1376	1453 9335	988 6748	672 2988	457 1632	310 8710	211 3922	143 7467	97 7477	66 4684	45 1985	30 7350	20 8998
69	4761	32 8509	2266 7121	1564 0313	1079 1816	744 6353	513 7983	354 5208	244 6194	168 7873	116 4632	80 3596	55 4481	38 2592	26 3988
70	4900	34 3000	2401 0000	1680 7000	1176 4900	0823 5430	0576 4801	0403 5360	0282 4752	0197 7326	0138 4128	0096 8890	0067 8223	0047 4756	0033 2329
71	5041	35 7911	2541 1681	1804 2293	1281 0028	909 5120	645 7535	458 4850	325 5243	231 1222	164 0968	116 5087	82 7212	58 7320	41 6997
72	5184	37 3248	2687 3856	1934 9176	1393 1406	1003 0613	722 2041	519 9869	374 3906	269 5612	194 0840	139 7405	100 6131	72 4415	52 1578
73	5329	38 9017	2839 8241	2073 0715	1513 3422	1104 7398	806 4600	588 7158	429 7625	313 7266	229 0204	167 1849	122 0450	89 0928	65 0377
74	5476	40 5224	2998 6576	2219 0066	1642 0649	1215 1280	899 1947	665 4041	492 3990	364 3752	269 6377	199 5319	147 6536	109 2636	80 8551
75	5625	42 1875	3164 0625	2373 0468	1779 7851	1334 8388	1001 1291	750 8468	563 1351	422 3513	316 7635	237 5726	178 1794	133 6346	100 2259
76	5776	43 8976	3336 2176	2535 5253	1926 9992	1464 5194	1113 0347	845 9064	642 8888	488 5955	371 3326	282 2127	214 4817	163 0061	123 8846
77	5929	45 6533	3515 3041	2706 7841	2084 2238	1604 8523	1235 7362	951 5169	732 6680	564 1543	434 3988	334 4871	257 5550	198 3174	152 7044
78	6084	47 4552	3701 5056	2887 1743	2251 9960	1756 5568	1370 1143	1068 6892	833 5775	650 1905	507 1486	395 5759	308 5492	240 6683	187 7213
79	6241	49 3039	3895 0081	3077 0563	2430 8745	1920 3908	1517 1088	1198 5159	946 8276	747 9938	590 9151	466 8229	368 7901	291 3441	230 1619
80	6400	51 2000	4096 0000	3276 8000	2621 4400	2097 1520	1677 7216	1342 1772	1073 7418	0858 9934	0687 1947	0549 7558	0439 8046	0351 8437	0281 4749
81	6561	53 1441	4304 6721	3486 7844	2824 2953	2287 6792	1853 0201	1500 9463	1215 7665	984 7709	797 6644	646 1081	523 3476	423 9115	343 3683
82	6724	55 1368	4521 2176	3707 3984	3040 0667	2492 8547	2044 1408	1676 1955	1374 4803	1127 0738	924 2005	757 8444	621 4324	509 5746	417 8511
83	6889	57 1787	4745 8321	3939 0406	3269 4037	2713 6050	2252 2922	1869 4025	1551 6041	1287 8314	1068 9000	887 1870	736 3652	611 1831	507 2820
84	7056	59 2704	4978 7136	4182 1194	3512 9803	2950 9034	2478 7589	2082 1574	1749 0122	1469 1703	1234 1030	1036 6465	870 7831	731 4578	614 4245
85	7225	61 4125	5220 0625	4437 0531	3771 4951	3205 7708	2724 9052	2316 1694	1968 7440	1673 4324	1422 4175	1209 0549	1027 6966	873 5421	742 5108
86	7396	63 6056	5470 0816	4704 2701	4045 6723	3479 2782	2992 1792	2573 2741	2213 0157	1903 1935	1636 7464	1407 6019	1210 5376	1041 0624	895 3136
87	7569	65 8503	5728 9761	4984 2092	4336 2620	3772 5479	3282 1167	2855 4415	2484 2341	2161 2837	1880 3168	1635 8756	1423 2118	1238 1942	1077 2290
88	7744	68 1472	5996 9536	5277 3191	4644 0408	4086 7559	3596 3452	3164 7838	2785 0097	2450 8085	2156 7115	1897 9061	1670 1574	1469 7385	1293 3699
89	7921	70 4969	6274 2241	5584 0594	4969 8129	4423 1334	3936 5888	3503 5640	3118 1719	2775 1730	2469 9040	2198 2145	1956 4109	1741 2057	1549 6731
90	8100	72 9000	6561 0000	5904 9000	5314 4100	4782 9690	4304 6721	3874 2048	3486 7844	3138 1059	2824 2953	2541 8658	2287 6792	2058 9113	1853 0201
91	8281	75 3571	6857 4961	6240 3214	5678 6925	5167 6101	4702 5252	4279 2980	3894 1611	3543 6866	3224 7548	2934 5269	2670 4195	2430 0817	2211 3743
92	8464	77 8688	7163 9296	6590 8152	6063 5500	5578 4660	5132 1887	4721 6136	4343 8845	3996 3737	3676 6638	3382 5307	3111 9283	2862 9740	2633 9361
93	8649	80 4357	7480 5201	6956 8836	6469 9018	6017 0087	5595 8180	5204 1108	4839 8230	4501 0354	4185 9629	3892 9455	3620 4393	3367 0086	3131 3180
94	8836	83 0584	7807 4896	7339 0402	6898 6978	6484 7759	6095 6893	5729 9480	5386 1511	5062 9820	4759 2031	4473 6509	4205 2319	3952 9179	3715 7429
95	9025	85 7375	8145 0625	7737 8093	7350 9189	6983 3729	6634 2043	6302 4940	5987 3693	5688 0009	5403 6008	5133 4208	4876 7497	4632 9123	4401 2666
96	9216	88 4736	8493 4656	8153 7269	7827 5778	7514 4747	7213 8957	6925 3399	6648 3263	6382 3933	6127 0975	5882 0136	5646 7331	5420 8637	5204 0292
97	9409	91 2673	8852 9281	8587 3402	8329 7200	8079 8284	7837 4335	7602 3105	7374 2412	7153 0140	6938 4236	6730 2709	6528 3627	6332 5118	6142 5365
98	9604	94 1192	9223 6816	9039 2079	8858 4238	8681 2553	8507 6302	8337 4776	8170 7280	8007 3135	7847 1672	7690 2238	7536 4194	7385 6910	7237 9772
99	9801	97 0299	9605 9601	9509 9004	9414 8014	9320 6534	9227 4469	9135 1724	9043 8207	8953 3825	8863 8487	8775 2102	8687 4581	8600 5835	8514 5777

for Numbers 10 to 99

In cam calculations, numbers 10 to 99 would be used as 0.10 to 0.99. Decimal points would then be located to the left of the first digit. That is, $0.10^5 = 0.00001$, $0.99^{30} = 0.73970037$, etc. Thus, values provide calculation accuracy to eight places. Where no entries appear, ciphers occur in the first eight places when the decimal point precedes the numbers.

No.	17	18	19	20	21	22	23	24	25	26	27	28	29	30	No.
10															10
11															11
12															12
13															13
14															14
15															15
16															16
17															17
18															18
19															19
20															20
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24															24
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26															26
27															27
28															28
29															29
30															30
31															31
32															32
33															33
34	1														34
35	1														35
36	2	1													36
37	4	1													37
38	7	2	1												38
39	11	4	1												39
40	0000 0017	0000 0006	0000 0002	0000 0001											40
41	26	10	4	1											41
42	39	16	6	2	1										42
43	58	25	10	4	2										43
44	86	38	16	7	3	1									44
45	127	57	25	11	5	2	1								45
46	184	85	39	17	8	3	1								46
47	266	125	58	27	13	6	2	1							47
48	381	182	87	42	20	9	4	2	1						48
49	541	265	129	63	31	15	7	3	1						49
50	0000 0762	0000 0381	0000 0190	0000 0095	0000 0047	0000 0023	0000 0011	0000 0005	0000 0002	0000 0001					50
51	1068	544	277	141	72	36	18	9	4	2	1				51
52	1486	772	401	208	108	56	29	15	7	4	2	1			52
53	2054	1088	577	305	162	85	45	24	12	6	3	1	1		53
54	2822	1524	823	444	240	129	69	37	20	11	5	3	1		54
55	3856	2120	1166	641	352	194	106	58	32	17	9	5	2	1	55
56	5238	2933	1642	919	515	288	161	90	50	28	15	8	4	2	56
57	7077	4034	2299	1310	747	425	242	138	78	44	25	14	8	4	57
58	9512	5517	3199	1855	1076	624	362	210	121	70	40	23	13	7	58
59	1 2719	7504	4427	2612	1541	909	536	316	186	110	65	38	22	13	59
60	0001 6926	0001 0155	0000 6093	0000 3656	0000 2193	0000 1316	0000 0789	0000 0473	0000 0284	0000 0170	0000 0102	0000 0061	0000 0036	0000 0022	60
61	2 2418	1 3675	8341	5088	3104	1893	1155	704	429	262	159	97	59	36	61
62	2 9556	1 8325	1 1361	7044	4367	2707	1678	1040	645	400	248	153	95	59	62
63	3 8796	2 4441	1 5398	9700	6111	3850	2425	1528	962	606	382	240	151	95	63
64	5 0706	3 2451	2 0769	1 3292	8507	5444	3484	2230	1427	913	584	374	239	153	64
65	6 5997	4 2898	2 7883	1 8124	1 1780	7657	4977	3235	2102	1366	888	577	375	244	65
66	8 5555	5 6466	3 7267	2 4596	1 6233	1 0714	7071	4667	3080	2033	1341	885	584	385	66
67	11 0476	7 4019	4 9593	3 3227	2 2262	1 4915	9993	6695	4486	3005	2013	1349	904	605	67
68	14 2118	9 6640	6 5715	4 4686	3 0386	2 0663	1 4050	9554	6497	4418	3004	2042	1389	944	68
69	18 2152	12 5685	8 6722	5 9838	4 1288	2 8489	1 9657	1 3563	9358	6457	4455	3074	2121	1463	69
70	0023 2630	0016 2841	0011 3988	0007 9792	0005 5854	0003 9098	0002 7368	0001 9158	0001 3410	0000 9387	0000 6571	0000 4599	0000 3219	0000 2253	70
71	29 6068	21 0208	14 9248	10 5966	7 5235	5 3417	3 7926	2 6927	1 9118	1 3574	9637	6842	4858	3449	71
72	37 5536	27 0386	19 4678	14 0168	10 0921	7 2663	5 2317	3 7668	2 7121	1 9527	1 4059	1 0123	7288	5247	72
73	47 4775	34 6586	25 3008	18 4695	13 4827	9 8424	7 1849	5 2450	3 8288	2 7950	2 0404	1 4894	1 0873	7937	73
74	59 8327	44 2762	32 7644	24 2456	17 9418	13 2769	9 8249	7 2704	5 3801	3 9812	2 9461	2 1801	1 6133	1 1938	74
75	75 1694	56 3771	42 2828	31 7121	23 7840	17 8380	13 3785	10 0339	7 5254	5 6440	4 2330	3 1747	2 3810	1 7858	75
76	94 1523	71 5557	54 3823	41 3306	31 4112	23 8725	18 1431	13 7887	10 4794	7 9644	6 0529	4 6002	3 4961	2 6570	76
77	117 5824	90 5384	69 7146	53 6802	41 3337	31 8270	24 5068	18 8702	14 5300	11 1881	8 6148	6 6334	5 1077	3 9329	77
78	146 4226	114 2096	89 0835	69 4851	54 1984	42 2747	32 9743	25 7199	20 0615	15 6480	12 2054	9 5202	7 4258	5 7921	78
79	181 8279	143 6440	113 4787	89 6482	70 8221	55 9494	44 2000	34 9180	27 5852	21 7923	17 2159	13 6006	10 7444	8 4881	79
80	0225 1799	0180 1439	0144 1151	0115 2921	0092 2337	0073 7869	0059 0295	0047 2236	0037 7789	0030 2231	0024 1785	0019 3428	0015 4742	0012 3794	80
81	278 1283	225 2839	182 4800	147 8088	119 7251	96 9773	78 5516	63 6268	51 5377	41 7455	33 8139	27 3892	22 1853	17 9701	81
82	342 6379	280 9631	230 3897	188 9196	154 9140	127 0295	104 1642	85 4146	70 0400	57 4328	47 0949	38 6178	31 6666	25 9666	82
83	421 0440	349 4665	290 0572	240 7475	199 8204	165 8509	137 6563	114 2547	94 8314	78 7100	65 3293	54 2233	45 0054	37 3544	83
84	516 1166	433 5379	364 1719	305 9043	256 9596	215 8461	181 3107	152 3010	127 9328	107 4636	90 2694	75 8263	63 6941	53 5030	84
85	631 1342	536 4640	455 9944	387 5953	329 4560	280 0376	238 0319	202 3271	171 9780	146 1813	124 2541	105 6160	89 7736	76 3075	85
86	769 9697	662 1739	569 4696	489 7438	421 1797	362 2145	311 5045	267 8939	230 3887	198 1343	170 3955	146 5401	126 0245	108 3810	86
87	937 1892	815 3546	709 3585	617 1419	536 9134	467 1147	406 3898	353 5591	307 5964	267 6089	232 8197	202 5531	176 2212	153 3125	87
88	1138 1655	1001 5856	881 3953	775 6279	682 5525	600 6462	528 5687	465 1404	409 3236	360 2047	316 9802	278 9425	245 4694	216 0131	88
89	1379 2090	1227 4960	1092 4715	972 2996	865 3466	770 1585	685 4411	610 0425	542 9379	483 2147	430 0611	382 7543	340 6514	303 1797	89
90	1667 7181	1500 9463	1350 8517	1215 7665	1094 1898	0984 7709	0886 2938	0797 6644	0717 8979	0646 1081	0581 4973	0523 3476	0471 0128	0423 9115	90
91	2012 3507	1831 2391	1666 4276	1516 4491	1379 9687	1255 7715	1142 7520	1039 9043	946 3130	861 1448	783 6417	713 1140	648 9337	590 5297	91
92	2423 2212	2229 3635	2051 0144	1886 9332	1735 9786	1597 1003	1469 3323	1351 7857	1243 6428	1144 1514	1052 6193	968 4097	890 9369	819 6620	92
93	2912 1257	2708 2769	2518 6975	2342 3887	2178 4215	2025 9320	1884 1167	1752 2286	1629 5726	1515 5025	1409 4173	1310 7581	1219 0050	1133 6747	93
94	3492 7983	3283 2304	3086 2366	2901 0624	2726 9986	2563 3787	2409 5760	2265 0014	2129 1013	2001 3552	1881 2739	1768 3975	1662 2936	1562 5560	94
95	4181 2033	3972 1431	3773 5360	3584 8592	3405 6162	3235 3354	3073 5686	2919 8902	2773 8957	2635 2009	2503 4408	2378 2688	2259 3554	2146 3876	95
96	4995 8680	4796 0333	4604 1920	4420 0243	4243 2233	4073 4944	3910 5546	3754 1324	3603 9671	3459 8084	3321 4161	3188 5594	3061 0171	2938 5764	96
97	5958 2604	5779 5126	5606 1272	5437 9434	5274 8051	5116 5609	4963 0641	4814 1722	4669 7470	4529 6546	4393 7650	4261 9520	4134 0934	4010 0706	97
98	7093 2176	6951 3533	6821 3262	6676 0797	6542 5581	6411 7069	6283 4728	6157 8033	6034 6472	5913 9543	5795 6752	5679 7617	5566 1665	5454 8431	98
99	8429 4319	8345 1376	8261 6862	8179 0693	8097 2786	8016 3058	7963 1428	7856 7814	7778 2135	7700 4314	7623 4271	7547 1928	7471 7209	7397 0037	99

The simplest polynomial satisfying these conditions is

$$y = C_0 + C_1x^1 + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5$$

which reduces to

$$y = 1 + C_2x^2 + C_4x^4 + C_5x^5$$

By use of the method given in TABLE 1, the coefficients can be evaluated and the final equations for displacement and its successive derivatives are given in *Fig.* 9. If dynamics of this cam form are to be closely controlled, *Fig.* 9 shows that with unequal positive and negative acceleration peaks, the maximum value is greater than desirable or necessary.

Maximum acceleration for the same initial conditions can be reduced by altering the powers of the polynomial and bringing the positive and negative peaks into near equivalence. In such a power change for this profile type, the second power should not be altered, for the term containing it defines acceleration at $x = 0$. Neither should any powers whose coefficients were found to be zero from the conditions be reinstated. Hence, in this case power changes must be confined to the last two terms.

As a trial, which proves to be satisfactory, let the powers 2-4-5 be replaced by 2-5-6; that is,

$$y = 1 + C_2x^2 + C_5x^5 + C_6x^6$$

The results are shown in *Fig.* 10. Here the maximum values for positive and negative acceleration are approximately equal (5.24 versus -5). Additionally, just as the absence of the third power yielded zero jerk and tangential acceleration at $x = 0$ in the original 2-4-5 equation, the absence also of the fourth power in the 2-5-6 equation yields a zero fourth derivative (not shown in *Fig.* 10) and tangential jerk at $x = 0$.

If tangency of acceleration to its horizontal axis is desired at $x = 1$, there should be added to the conditions: when $x = 1$, $y''' = 0$. Manipulation of the powers again would permit equalization of maximum positive and negative acceleration, though at a higher value than when jerk was not limited to zero at $x = 1$.

A demonstration of progressive changes in powers is given by *Fig.* 11 for a group of higher-order polynomials. The conditions for each of these curves are: When $x = 0$; $y = 0.350$, $y' = 0$, $y''' = 0$. When $x = 1$; $y = 0$, $y' = 0$, $y'' = 0$, $y''' = 0$, and $y^{iv} = 0$. These curves characterize the type successfully adapted to automotive cams.

Table 3—Coefficient Evaluations

(for cases 1-A, 1-B, 1-C, 2-A, and 2-B)

At terminal point (where $x = 0$):

$$C_0 = h_1,\ C_1 = \frac{a_1}{1!},\ C_2 = \frac{b_1}{2!},\ C_3 = \frac{c_1}{3!},\ \text{etc.},\ C_n = \frac{m_1}{n!}$$

where h_1 = lift and a_1, b_1, c_1, m_1 = assigned values for successive terminal derivatives of displacement.

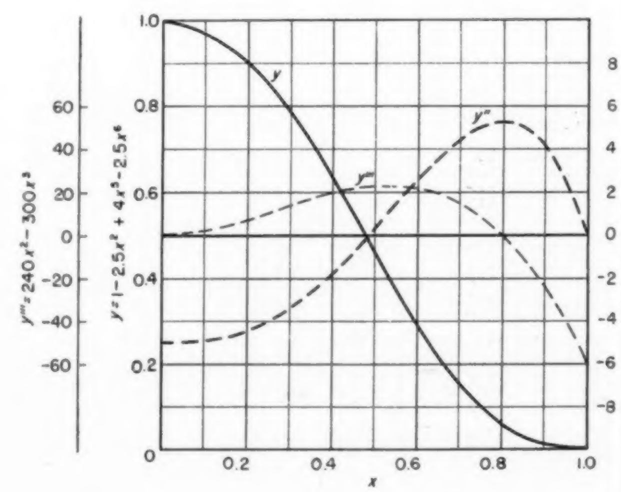
At initial point (where $x = 1$):

$$C_p = -\frac{qrs\cdots C_0}{(q-p)(r-p)(s-p)\cdots} - \frac{(q-1)(r-1)(s-1)\cdots a_1}{(q-p)(r-p)(s-p)\cdots} - \frac{(q-2)(r-2)(s-2)\cdots b_1}{2\,(q-p)(r-p)(s-p)\cdots} - \frac{(q-3)(r-3)(s-3)\cdots c_1}{(3)(2)(q-p)(r-p)(s-p)\cdots} - \cdots - \frac{(q-n)(r-n)(s-n)\cdots m_1}{n!\,(q-p)(r-p)(s-p)\cdots}$$

$$C_q = -\frac{prs\cdots C_0}{(p-q)(r-q)(s-q)\cdots} - \frac{(p-1)(r-1)(s-1)\cdots a_1}{(p-q)(r-q)(s-q)\cdots} - \frac{(p-2)(r-2)(s-2)\cdots b_1}{2\,(p-q)(r-q)(s-q)\cdots} - \frac{(p-3)(r-3)(s-3)\cdots c_1}{(3)(2)(p-q)(r-q)(s-q)\cdots} - \cdots - \frac{(p-n)(r-n)(s-n)\cdots m_1}{n!\,(p-q)(r-q)(s-q)\cdots}$$

C_r, C_s, etc., follow the same pattern shown for C_p and C_q. Power symbols p, q, r, etc., may arbitrarily represent any whole number power without regard for a strict sequential relationship.

Fig. 10—Effect of changing 2-4-5 polynomial of *Fig.* 9 to 2-5-6 to obtain near equivalence of maximum positive and negative acceleration



RRR Cams: Continuously intermittent action given by these cams might also be obtained by eccentrics or linkages with greater advantage in many applications. However, if such are desired, they can be specified in terms of a polynomial. If the rise and return portions of the action are to be equivalent, and to yield a continuous acceleration curve, the following conditions can be set: When $x = 0$; $y = 1$, $y' = 0$, $y''' = 0$, and when $x = 1$; $y = 0$, $y' = 0$, and $y''' = 0$. The basic polynomial is

$$y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5$$

which from the conditions reduces to

$$y = C_0 + C_2x^2 + C_4x^4 + C_5x^5$$

Upon evaluation of the coefficients by simultaneous equations, the expression for displacement is given in *Fig.* 12. The acceleration curve is symmetrical with maximum values of ±5. TABLE 1 could not be used in evaluating these coefficients because its use is limited to consecutive derivatives of zero value at $x = 1$.

If, for any reason, displacement or any other property is to be distorted, such as with a more rapid rise at the start of action than at the close, powers of the polynomial can be altered to produce such effects. As long as the same initial conditions are retained, acceleration will continue as a smooth though unsymmetrical curve.

Polynomial Boundary Conditions: Preceding discussion has shown how profiles for different types of cams may be developed in step-by-step fashion. As a corrolary, this design method can be restated in a more generalized form. Familiarity with the system from this point of view may facilitate its adaptation to different design circumstances. So far, at the beginning or end of the cam event, the only controls have been zero, or no control at all.

As is quite apparent in this development the polynomial equation applied to cams is based exclusively upon event boundary conditions. With the term "control" meaning the process of intentional assignment of boundary conditions, polynomials can be classified according to two attributes: location of control and degree of control.

Location implies:

1. Control of the terminal point (where $x = 0$, $y = 1$, etc.).
2. Control of the initial point (where $x = 1$, $y = 0$, etc.).

Three different degrees of control might be exercised at either of the foregoing control locations:

A. No control.
B. Assignment of zero to one or more displacement derivatives.
C. Assignment of *finite* quantities to one or more displacement derivatives.

These three different degrees of control usually prevail for different derivatives in most cam polynomials.

Terminal control might be considered first. With

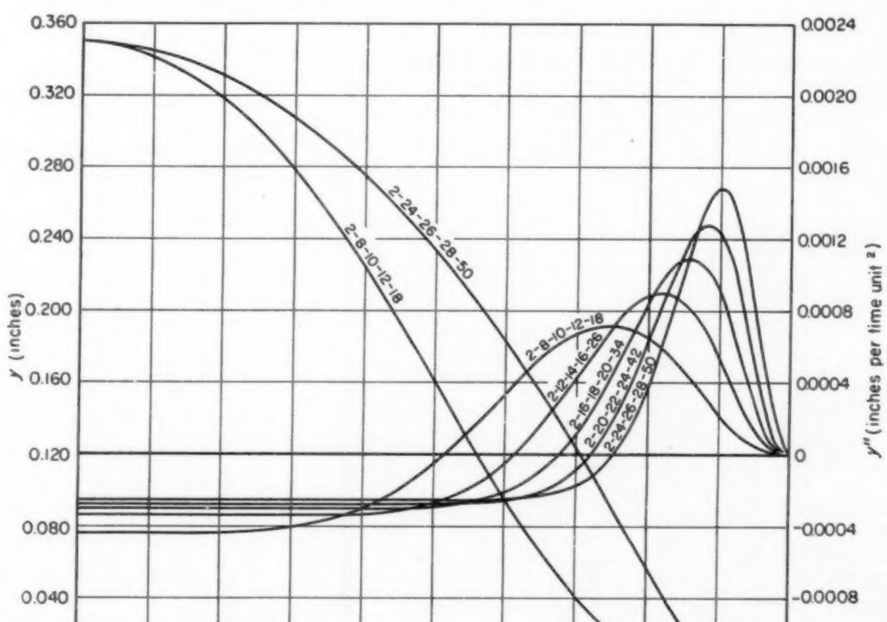


Fig. 11 — Effect of progressive power changes upon high-order polynomials of the DRRD type profile

no control the coefficients of x are unrestricted. If $y = h$, the expression for y must have a constant equal to h. But if $y' = 0$, coefficient of the x term in the expression for y must be zero. Similarly, if $y'' = 0$, the x^2 coefficient must be zero. This process can be carried on for any given number of derivatives desired. Conversely, if any given derivatives must not equal zero, the corresponding coefficient for the x-term at its proper power must not equal zero. These facts hold absolutely for conditions 1-A and 1-B.

For condition 1-C if $y' = a_1$, $y = \ldots + a_1x + \ldots$; if $y'' = b_1$, $y = \ldots + (b_1x^2/2) + \ldots$; if $y''' = c_1$, $y = \ldots + [c_1x^3/(3)(2)] + \ldots$ or, if y to the nth derivative equals m, $y = \ldots + (mx^n/n!) + \ldots$

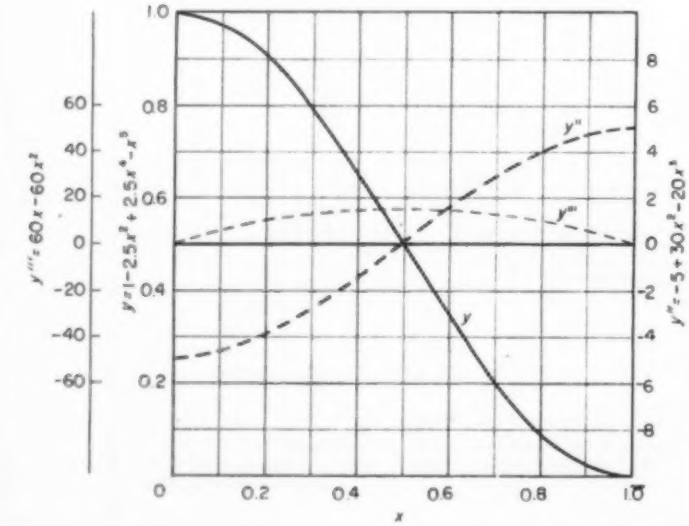
Now, controls can be considered at the initial point (where $x = 1$). Here controlled conditions for a specific power of x need not be assigned. Whether the first derivative of y or the nth derivative is being controlled, the coefficient may be applied to an x-term of any power, if that term is not already controlled by an assigned coefficient at either end of the event. That is, there must be as many terms in the polynomial for y as there are assigned controls, including zero coefficients. In rare cases, controls on both ends may be provided by a single term. But generally, only when no control is required at the terminal for a given derivative may the corresponding power term be used for a control at the initial point.

For the case 2-B, the values for the coefficients are readily established no matter how many derivatives are required. Let the polynomial have the general form:

$$y = h + C_px^p + C_qx^q + C_rx^r + C_sx^s + \ldots$$

where $C_0 = h$, the lift. Then, the coefficients can be evaluated according to the equations earlier given in TABLE 1. Incidentally, these coefficient equations, as well as those of the same type to follow, have been derived by methods ordinarily employed in the solution of simultaneous equations. Direct manipulation and substitution proved more facile than the method of determinants, despite the obvious laboriousness for multiterm polynominals.

Fig. 12—Basic polynomial for RRR type profile



Next, for cases 1-A, 1-B, 1-C, 2-A, and 2-B (only case 1-C added to the cases just discussed), the coefficient equations are given in TABLE 3. These equations in this table, as well as those in TABLE 1, hold for any number of terms and for assignments of any controls, exclusive of case 2-C. This flexibility diminishes when case 2-C is encountered. No "master" coefficient equations can be set up that apply universally. Instead, when the control circumstances are assigned, coefficient equations can readily be developed for those particular conditions.

For example, if two power terms (p and q) apply and $y' = a_0$ at $x = 1$, then

$$C_p = -\frac{q\,C_0}{(q-p)} - \frac{a_0}{(q-p)} - \frac{(q-1)a_1}{(q-p)} - \frac{(q-2)b_1}{2\,(q-p)} - \frac{(q-3)c_1}{(2)(3)(q-p)} - \ldots$$

$$C_q = \frac{p\,C_0}{(p-q)} - \frac{a_0}{(p-q)} - \frac{(p-2)b_1}{2(p-q)} - \frac{(p-3)c_1}{(2)(3)(p-q)} - \ldots$$

Further, if three power terms (p, q, and r) apply and if $y' = a_0$ and $y'' = b_0$ at $x = 1$, one coefficient will be

$$C_p = -\frac{qr\,C_0}{(q-p)(r-p)} - \frac{(r-q-1)a_0}{(q-p)(r-p)} - \frac{(q-1)(r-1)a_1}{(q-p)(r-p)} + \frac{b_0}{(q-p)(r-p)} - \frac{(q-2)(r-2)\,b_1}{2\,(q-p)(r-p)} + \frac{(q-3)(r-3)\,c_1}{(3)(2)(q-p)(r-p)}$$

Next, if four terms are used with $y' = a_0$, $y'' = b_0$, and $y''' = c_0$ at $x = 1$,

$$C_p = -\frac{qrs\,C_0}{(q-p)(r-p)(s-p)} - \frac{(sr+sq-s+rq-r-q+1)a_0}{(q-p)(r-p)(s-p)} - \frac{(q-1)(r-1)(s-1)a_1}{(q-p)(r-p)(s-p)} + \frac{(s+r+q-3)\,b_0}{(q-p)(r-p)(s-p)} - \frac{(q-2)(r-2)(s-2)\,b_1}{2\,(q-p)(r-p)(s-p)} - \frac{c_0}{(q-p)(r-p)(s-p)} - \frac{(q-3)(r-3)(s-3)\,c_1}{2\,(3)(q-p)(r-p)(s-p)}$$

From here on the pattern becomes complicated and cumbersome. With this type of equation, finite values, or zeros, may be used for a_1, b_1, c_1, and a_0, b_0, c_0.

Three-Stage Example: Mathematical principles so far developed can be demonstrated by a hypothetical

design based on different types of controls. Suppose that a cam is to be designed with a total lift of 2.3 units over a 20-increment range and with three distinct stages of action as follows:

1. A rapidly starting rise over 10 increments to a lift of 1 unit; starting with zero velocity, acceleration and jerk; and finishing with a velocity of 0.5 unit per 10 increments, and zero acceleration and jerk.
2. A constant velocity of 0.5 units per 10 increments over a 2-increment range (lift = 0.1.)
3. A slowly starting rise over 8 increments to a lift of 1.2 unit; starting with a velocity of 0.5 unit per 10 increments, zero acceleration and jerk; and ending with an acceleration of −0.3 unit per 10 increments squared, and zero velocity and jerk.

Expressed in symbols, these conditions are:

Stage 1	Lift	Velocity	Acc.	Jerk	Range
Initial	$h_0 = 0$	$a_0 = 0$	$b_0 = 0$	$c_0 = 0$	$x_r = 1$
Terminal	$h_1 = 1$	$a_1 = -0.5$	$b_1 = 0$	$c_1 = 0$	
Stage 2					
Initial	$h_0 = 0$	$a_0 = -0.5$	$b_0 = 0$	$c_0 = 0$	$x_r = 0.2$
Terminal	$h_1 = 0.1$	$a_1 = -0.5$	$b_1 = 0$	$c_1 = 0$	
Stage 3					
Initial	$h_0 = 0$	$a_0 = -0.5$	$b_0 = 0$	$c_0 = 0$	$x_r = 0.8$
Terminal	$h_1 = 1.2$	$a_1 = 0$	$b_1 = -0.3$	$c_1 = 0$	

STAGE 1: From the tabulation of conditions it is obvious that the eight conditions for stage 1 require a polynomial of eight terms. Furthermore, it will be recalled that terminal conditions automatically dictate the assignment of specific powers: 0, 1, 2, etc. And, initial conditions permit the arbitrary assignment of any powers to the remaining x-terms. Therefore, the general equation for stage 1 is

$$_1y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_px^p + C_qx^q + C_rx^r + C_sx^s$$

From TABLE 3 and the assigned conditions, $C_0 = h_1 = 1$, $C_1 = a_1 = -0.5$, $C_2 = b_1/2 = 0$, $C_3 = c_1/(3)(2) = 0$. Next, for a rather rapid starting rise, let the remaining powers be: $p = 6$, $q = 8$, $r = 10$, $s = 12$. Then C_p, for example, can be calculated from TABLE 3:

$$C_p = -\frac{(8)(10)(12)(1)}{(2)(4)(6)} - \frac{(7)(9)(11)(-0.5)}{(2)(4)(6)}$$
$$= -20 + 7.21875 = -12.78125$$

Only two terms of the equation for C_p are involved because the remaining terms contain b_1, c_1, etc., which are equal to zero. Applications of the remaining coefficient equations of TABLE 3 lead finally to the equations for stage 1:

$$_1y = 1 - 0.5 \times - 12.78\, x^6 + 29.53\, x^8 - 23.97\, x^{10} + 6.72\, x^{12}$$

$$_1y' = -0.5 - 76.69\, x^5 + 236.25\, x^7 - 239.69\, x^9 + 80.63\, x^{11}$$

$$_1y'' = -383.44\, x^4 + 1653.75\, x^6 - 2157.19\, x^8 + 886.88\, x^{10}$$

$$_1y''' = -1533.75\, x^3 + 9922.5\, x^5 - 17{,}257.5\, x^7 + 8868.75\, x^9$$

A quick check of the equations against the assigned

conditions is advisable at this stage. Such a verification can be conducted visually since displacement and its derivatives are to be checked at $x = 1$ and $x = 0$. Values can be calculated with the aid of powers given in TABLE 2, and resulting curves can be plotted, *Fig.* 13.

STAGE 2: Since stage 2 is simply a constant-velocity portion, no calculations are required. Because conditions at the terminal point of stage 1 and the initial point of stage 2 were assigned the same values, the plotted curves are continuous, *Fig.* 13.

STAGE 3: A somewhat different problem is encountered in the third stage. In stage 1 the derivatives are based on 10 increments totalling a range of 1. If the various functions are to be continuous, the 8-increment base of stage 3, resulting with a range of 0.8, must be adjusted. If any term without correction were C_nx^n, then the corrected term is $C_n'x^n = C_nx^n/x_r^n$ or, $C_n' = C^n/x_r^n$. This compensation is universally required. It had not been demonstrated for stage 1 since $({}_1x_r)^n = (1)^n = 1$. Applying this adjustment to stage 3 gives $C_n' = C_n/(0.8)^n$.

Next, the remaining powers must be selected for stage 3 consistent with the conditions prescribed. For lowest possible acceleration, the following powers are chosen: ${}_3p = 4$, ${}_3q = 5$, ${}_3r = 6$, ${}_3s = 7$. Then, from TABLE 3,

$$C_2' = \frac{C_2}{({}_3x_r)^2} = \frac{1}{({}_3x_r)^2}\left[\frac{b_1}{2!}\right] =$$
$$\frac{1}{(0.8)^2}\left[\frac{-0.3}{(1)(2)}\right] = \frac{-0.15}{(0.8)^2}$$

Next, coefficients for the 4,5,6 and 7 power terms (initial conditions) can be obtained from the generalized equation developed in the foregoing for case 2-C. Because a_1, b_0, c_0 and c_1 all equal zero by assignment, this earlier equation reduces to the form:

$$C_p = -\frac{qrs\,C_0}{(q-p)(r-p)(s-p)} - \frac{(sr + sq - s + rq - r - q + 1)a_0}{(q-p)(r-p)(s-p)} - \frac{(q-2)(r-2)(s-2)\,b_1}{2\,(q-p)(r-p)(s-p)}$$

Then, with range adjustment taken into account, this equation gives

$$C_4' = \frac{C_4}{({}_3x_r)^4} = \frac{1}{(0.8)^4}\left\{-\frac{(5)(6)(7)(1.2)}{(1)(2)(3)} - \frac{[(7)(6)+(7)(5)-7+(6)(5)-6-5+1](-0.5)}{(1)(2)(3)} - \frac{(3)(4)(5)(-0.3)}{(2)(1)(2)(3)}\right\}$$
$$= \frac{1}{(0.8)^4}(-42 + 7.5 + 1.5) = \frac{-33}{(0.8)^4}$$

Coefficients C_5', C_6' and C_7' are found in the same

fashion. The foregoing shortened equation for C_p $(p=4)$ can be rewritten for C_q $(q=5)$ by interchanging the symbols p and q. Or,

$$C_q = \frac{prs\,C_0}{(p-q)(r-q)(s-q)} - \frac{(sr+sp-s+rp)}{(p-q)(r-q)(s-q)} \text{ etc.}$$

This same process applies to the evaluation of C_6' and C_7'.

Finally, the coefficients in summary are

$$C_2 = -0.15\,/(0.8)^2 = -0.23438$$

$$C_4 = -33.0\,/(0.8)^4 = -80.56641$$

$$C_5 = +78.3\,/(0.8)^5 = +238.95264$$

$$C_6 = -64.75\,/(0.8)^6 = -247.00165$$

$$C_7 = +18.4\,/(0.8)^7 = +87.73804$$

The displacement equation is

$$_3y = 2.3 - 0.23438\,x^2 - 80.56641\,x^4 + 238.95264\,x^5 - 247.00156\,x^6 + 87.73804\,x^7$$

Obtaining derivatives of the displacement equation requires that each successive derivative must also be adjusted with respect to the range. That is,

$$d\,(C_n'x^n) = d\,[C_n\,(x/x_r)^n] = n\,C_n\,(x/x_r)^{n-1} = n\,(C_n/x_r^{n-1})\,x^{n-1}$$

But the actual term in the displacement equation is C_n/x_r^n, not C_n/x_r^{n-1}. Since $x_r\,(C_n/x_r^n) = C_n/x_r^{n-1}$,

$$d\,(C_n'\,x^n) = nx_r\,(C_n/x_r^n)\,x^{n-1}$$

This result means that in the process of differentia-

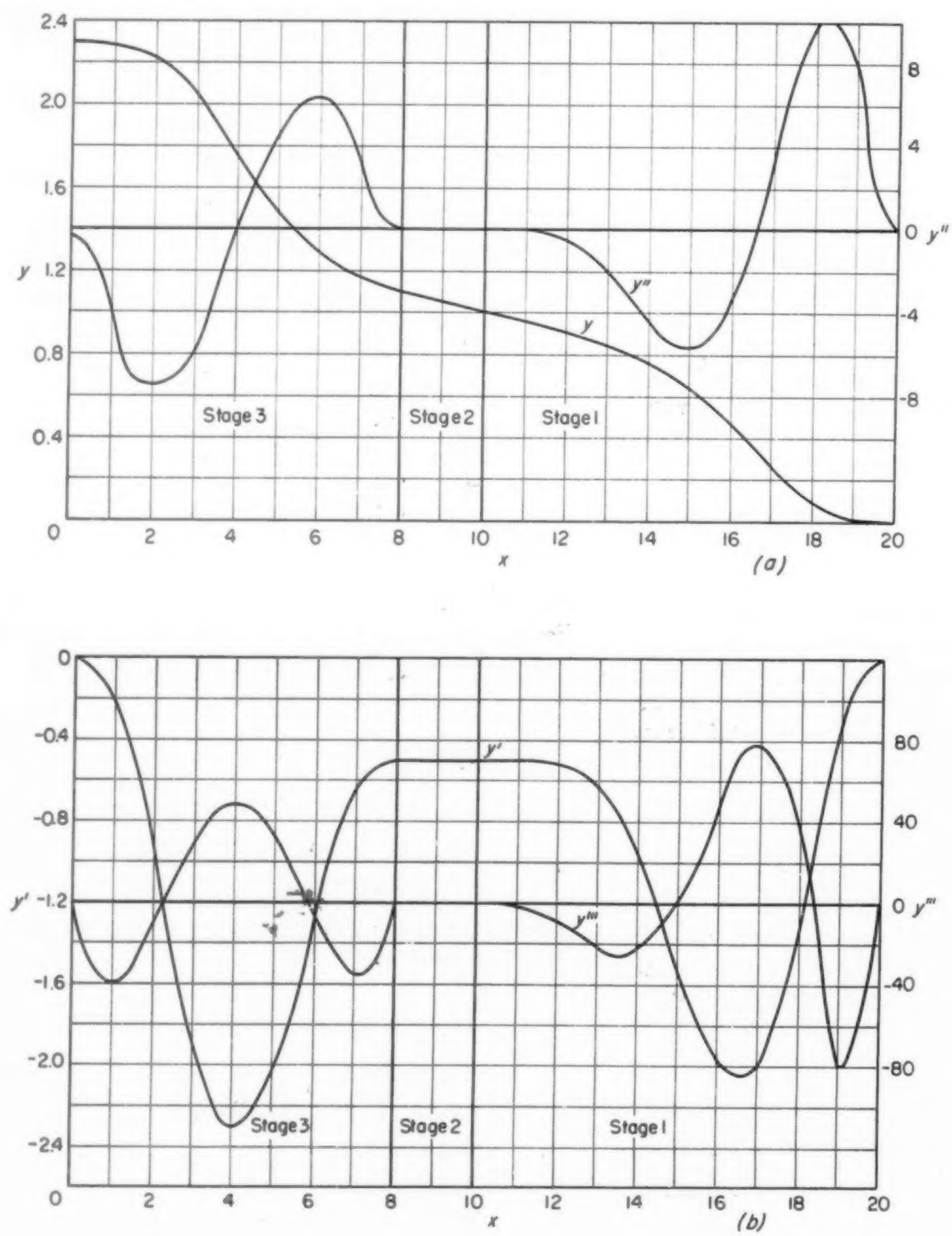


Fig. 13—Three stage example with displacement and acceleration plotted at *(a)*, velocity and jerk at *(b)*

tion the existing coefficient must be multiplied by the range x_r as well as by the power n of x.

Then by differentiation of the displacement equation, velocity is

$$_3y' = -0.375\,x - 257.8125\,x^3 + 955.8105\,x^4 - 1185.6079\,x^5 + 491.3330\,x^6$$

Again for acceleration the x_r multiplier is effective and

$$_3y'' = -0.3 - 618.75\,x^2 + 3058.5937\,x^3 - 4742.4317\,x^4 + 2358.3984\,x^5$$

Similarly, jerk is

$$_3y''' = -990\,x + 7340.625\,x^2 - 15{,}175.7814\,x^3 + 9433.5938\,x^4$$

Plots of these equations are shown in *Fig.* 13.

Summary: Applying polynomial equations to the development of cam contours is analogous to the process of curve fitting with polynominal equations —finding the polynominal equation of a curve that passes through a certain group of points. But in polydyne cam design, the points to be matched by the equated curve are related to more than the displacement curve. Some of the points are also effective for the successive derivatives of displacement.

The method provides for the matching of prescribed conditions of both terminals of a cam event by a single polynominal equation, or subevents of different natures, each with its own mathematical function, may be readily joined. Portions of curves between prescribed terminal conditions can be adjusted by selection of powers.

For simple conditions, writing the general displacement equation and its derivatives, inserting the prescribed conditions, and solving simultaneously yield final working equations with little trouble. For more involved conditions, with many unknown coefficients, use of generalized formulations such as those discussed in the latter part of this article is almost mandatory.

Contours so developed can be directly applied to the cam in any system where the motion of the mass at the end of the follower system strictly adheres to the character of the follower motion at the cam. But when flexibility, clearance, or mass of pertinent elements deny such adherence, the cam contour should be modified to accommodate the effects of these influences and impart to the end mass of the follower the desired motion without distortion. How to analyze cam and follower systems for these disturbing influences and account for them in cam design will be discussed in the next part of this series.

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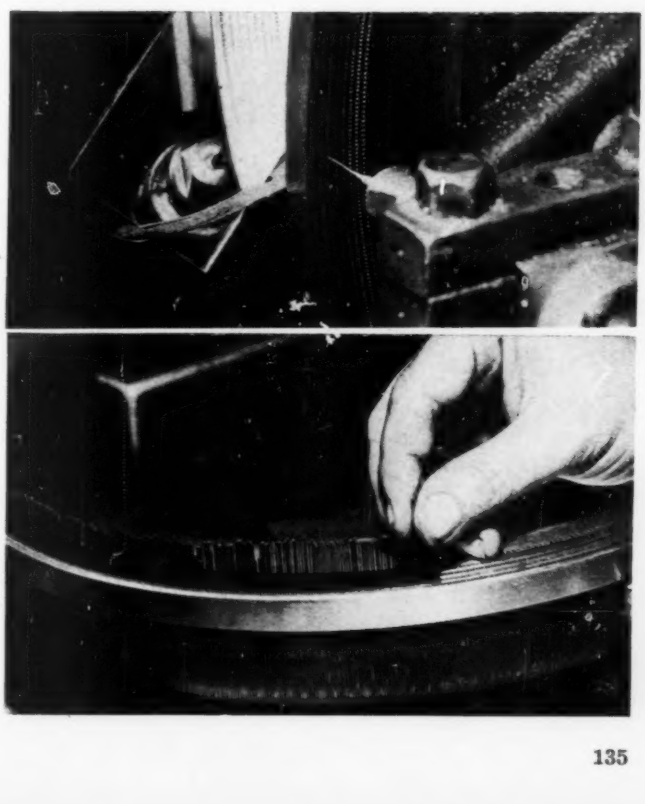
Seamless Tubing Aligns Teeth

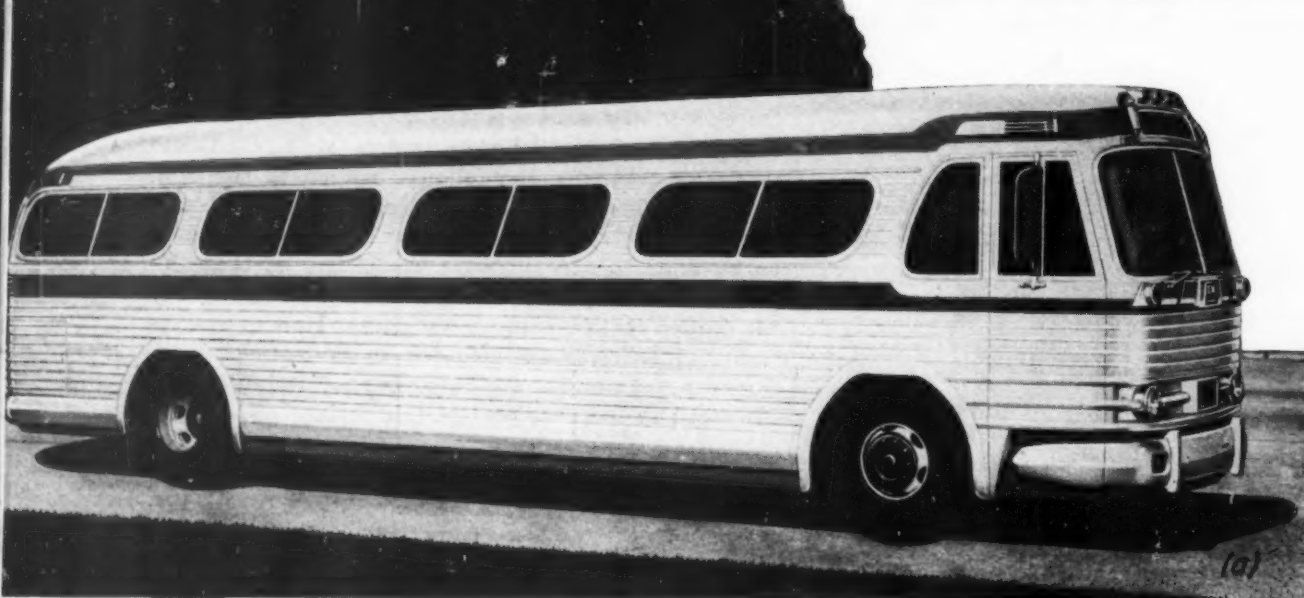
ALIGNMENT of spline teeth in comb circles manufactured for the worsted yarn industry is an interesting example of precision hand assembly. Comb circles for the worsted yarn industry are heavy cast-alloy rings through which a series of concentric circles of holes are bored, each row of holes being drilled and blanked to carry teeth of a different cross-sectional shape. At the R. H. Hood Co., spring-steel spline teeth are inserted through the ring to form a circular comb with multiple rows of specially shaped teeth.

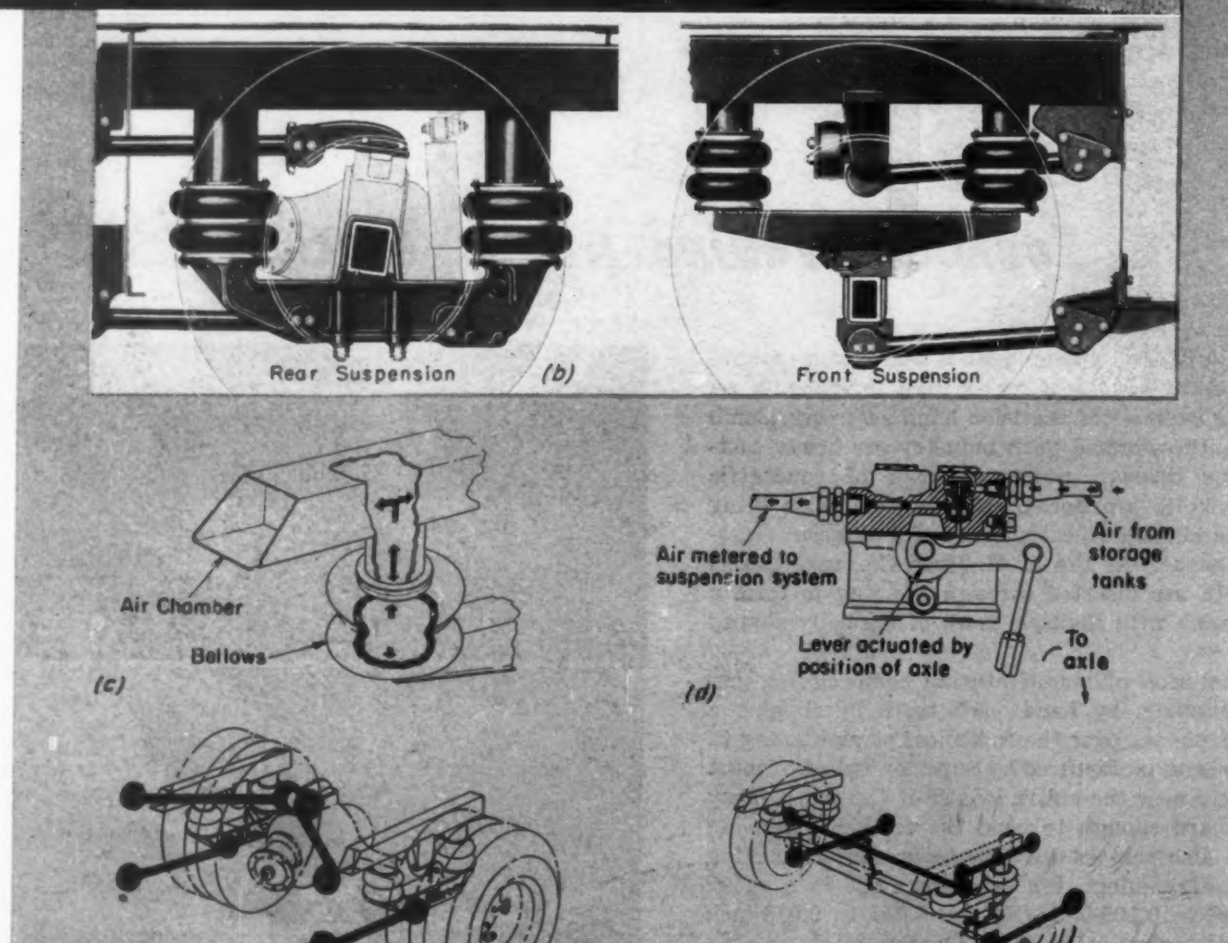


The final step of manufacturing comb circles consists of aligning, by hand, each tooth by slipping a preshaped pin set over the tooth and straightening it. The pin sets, manufactured by Superior Tube Co., must fit perfectly over the entire length of the tooth. They must be hard enough to bend the teeth for aligning and must also possess ductility to prevent fracturing during straightening. Pin sets are made in a range from 0.026 to 0.105-inch OD and 0.017 to 0.023-inch ID of high-carbon AISI E-52100 alloy steel.

Advantages claimed for the seamless tube pin sets are speed and accuracy, effecting savings in man-hours and reducing the number of production rejects.

CONTEM
DESIGN





ORARY

Bellows Replace Springs in New Coach

FLEXIBLE air bellows made of rubberized nylon tire fabric replace conventional metal leaf springs in a new General Motors coach, *a*. The coach body is suspended on eight bellows, two on each wheel, *b*. Compressed air is metered to an air chamber, consisting of a rectangular, rubber-lined sheet-steel box, and the connected bellows, *c*, by valves actuated by changes in position of the axle relative to the coach body, *d*. As the coach is loaded, air pressure from the coach storage tanks is automatically increased to keep the body at its original level. Direct, double-acting aircraft shock absorbers are used on both sides of the front and rear axles to control rebound and further cushion impact, while rubber-mounted radius rods, *e* and *f*, prevent lateral, longitudinal and torsional movement of each axle.

Air suspension accomplishes two purposes: (1) permits response to faint vibrations, especially in the higher frequencies where metal leaf springs fail to flex because of friction between leaves, and (2) makes possible a progressive rate of deflection, unlike leaf springs which deflect at a uniform rate. Thus, all kinds of road shock and vibration, from simple tire-tread vibration to heavy impact from deep chuck holes, can be absorbed.

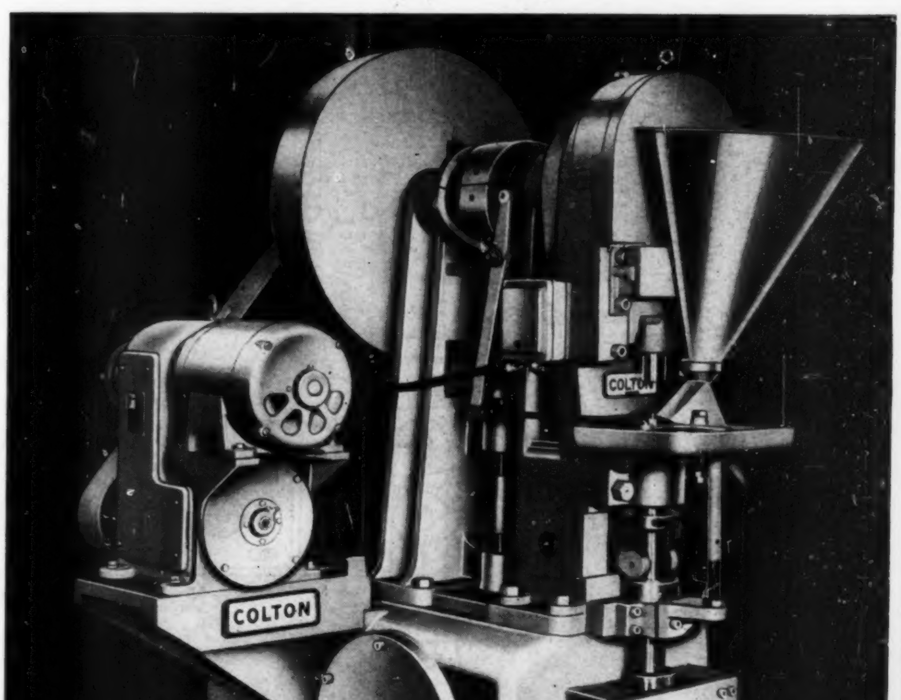
Among other features of the new coach are large "picture" windows, measuring 20 by 72 inches and the one-level floor with a minimum of wheelhouse projection through it. A full load of 41 seated passengers, each with 45 pounds of baggage, plus the driver and 100 gallons of fuel result in a rear-axle loading of 18,000 pounds.

Briquetting Press Has Double Compression Stroke

TO EQUALIZE density in "green" powder-metal parts, upper and lower rams of the briquetting press, right, both travel through a compression stroke. Since the die table "floats" between the two punches, a definite, adjustable ratio is established between top and bottom pressure for compacting metal powder briquets with uniform density.

Having a total stroke of 5 inches and a compression stroke of 1 inch, the upper punch, bottom next page, is operated by a crank. Stroke is adjusted by turning an eccentric sleeve on the crank pin, through a worm and worm-wheel arrangement accessible from the front of the machine.

The lower punch has a maximum compression stroke of 1 inch with an additional ejector stroke of 3 inches. Driven through two sets of helical bevel gears, a cam operates a toggle mechanism, the upper arm of which moves a slide. This compression slide always makes a 1-inch stroke, but actual compression stroke is adjusted by means of a compression-adjustment nut and sleeve which changes position of the

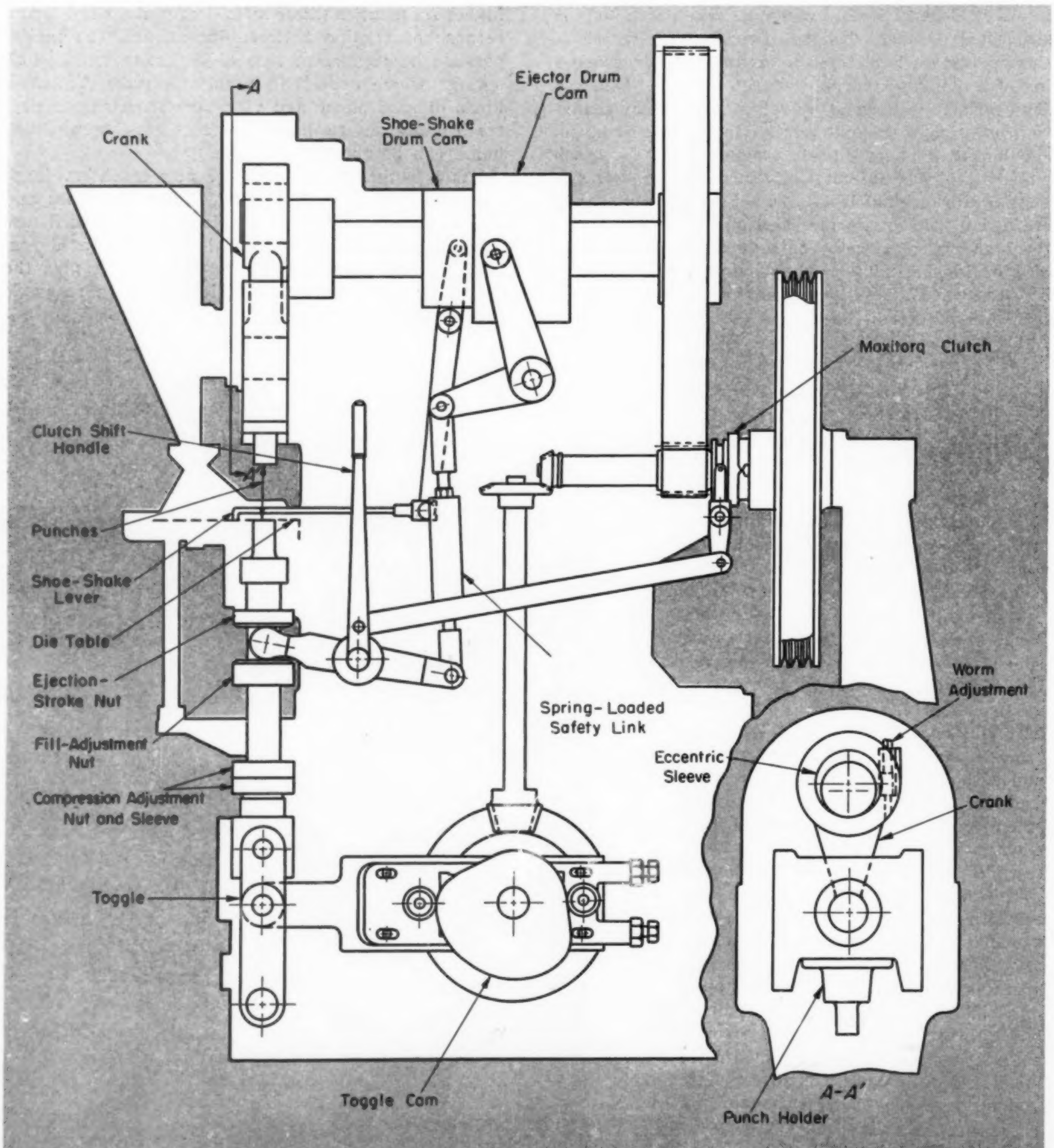


ram to vary the compression.

First, the amount of fill is determined by moving a fill-adjustment nut on the ram to rest against a bracket, and so limit the downward stroke. From this position, the compression-adjustment nut and sleeve are varied to give the amount of compression stroke needed. Thus, if the nut is 1/4-inch away from the compression slide when both are in their lowest positions, a 3/4-inch stroke will result.

The ejector rocker arm, operated from a drum cam, moves up against an adjustable nut which controls stroke of the ejection, and down against the top of the fill-adjustment nut to move the punch back to its original position. A safety, consisting of a spring-loaded link, keeps the ejector from breaking should a punch be locked in the up position.

Having a capacity of 35 tons on both punches, the No. 360 press built by Arthur Colton Co. uses a 7½ hp Reeves variable-speed unit with a V-belt drive to the flywheel. A double Maxitorq clutch acts as both a brake and as an adjustable-torque drive to the briquetting press.

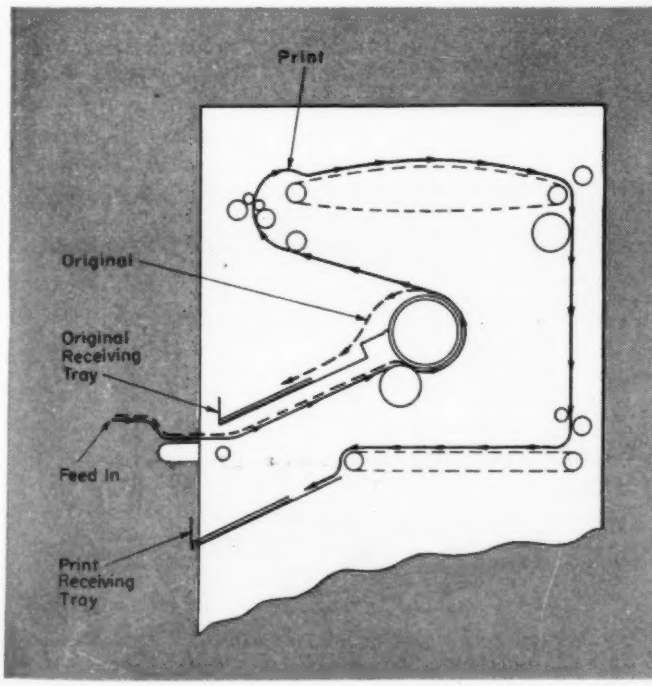


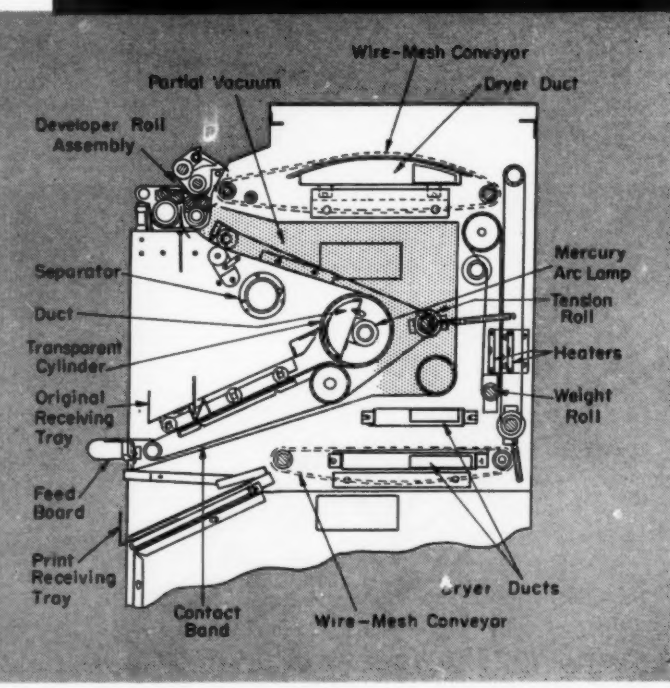
Vacuum-Pressure System Handles Large Reproductions

LARGE prints up to 20 inches wide are conveyed by an ingenious combination of conveyor belts and a pressure-vacuum system in the Copyflex 14 reproduction machine, right. Sensitized paper and original are fed into the machine, below left, and carried by cotton contact bands around a transparent cylinder containing a 1400-watt mercury arc quartz lamp, below right. The exposed copy print is then held to the under side of the contact bands by a partial vacuum inside the printer cabinet. A stream of air from a narrow slot along the length of a tubular separator unit is, at the same time, directed against the bottom side of the original. Thus, a low-pressure area created by the resulting Venturi effect on the under side of the original separates the original from the copy sheet, permitting the original to fall into a receiving tray.

After passing through the developer roll assembly, which meters the amount of developer applied, the print is conveyed through a dryer consisting of upper and lower horizontal wire-mesh conveyors and two sets of bands at the rear of the machine. Outside air, drawn into a duct running through the transparent cylinder surrounding the lamp, is evenly distributed along the length of the cylinder through holes in the duct, thus simultaneously cooling the outside cylinder surface in contact with the paper and picking up heat. This heated air is directed through dryer ducts at both sides of the print. Two 300-watt Calrod heaters warm the dryer contact bands.

A centrifugal blower operating at 800 rpm, rather than the conventional pressure-type fan, was found by Charles Bruning Co. Inc. to provide the required flow of 500 cfm of air at a static pressure of $\frac{1}{4}$-inch of water without operating at an excessive noise level. Air intake to the blower provides the partial vacuum in the printer chamber and draws air through the dryer ducts, and the exhaust supplies air to the separator through flexible hose.

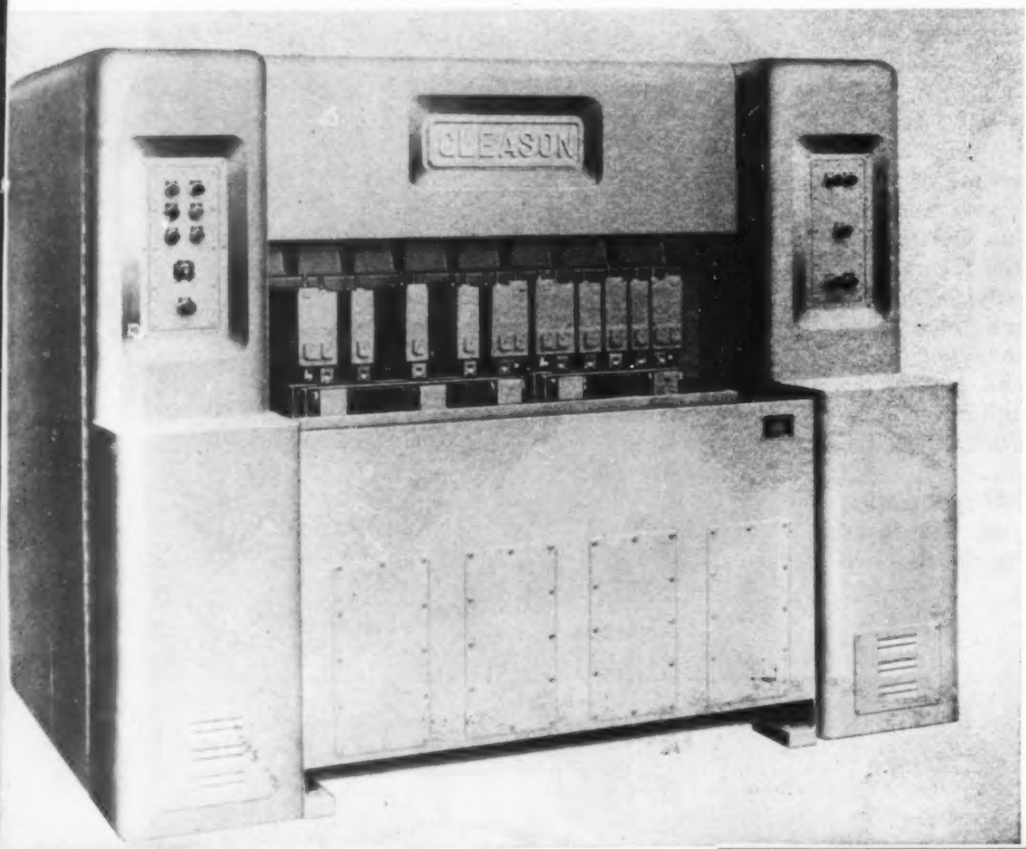




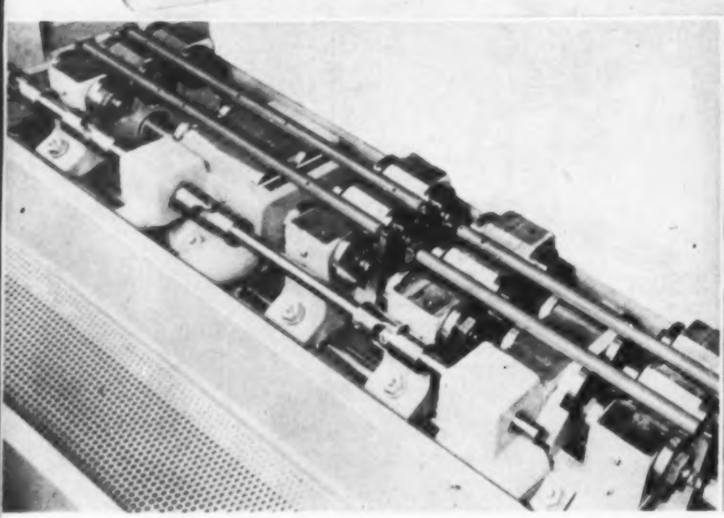
Quenching Machine Automatically Straightens Shafts

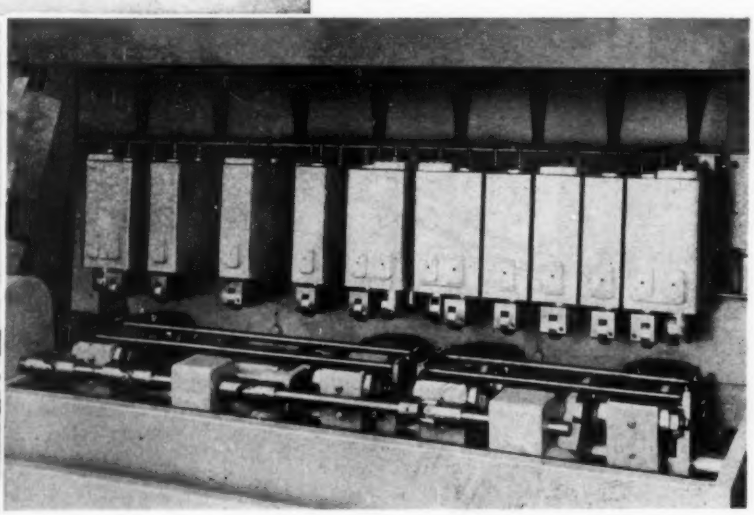
HOT BARS and shafts are automatically held straight and concentric while going through a controlled cycle in a new Gleason Works quenching machine, below. The heated part is placed on the lower rollers, bottom left, and the completely automatic cycle is started. The upper pressure-roller holder, bottom right, moves down, closing a guard and placing a positive preload on the part to guarantee exact positioning. Lower drive rollers rotate the part, and as it rotates a controlled volume of quenching fluid is directed at the end of the part, across the part, or in both directions at the same time.

The bar or shaft is held straight by securing the ends of the part between two lower drive rollers and a single upper pressure roller, equally spaced around the part. A forced straightening deflection is applied at several points along the shaft by raising and lowering idler rollers with a rack and gear arrangement driven by a shaft incorporating universal joints (photo at left). Pairs of lower drive rollers help rotate the part, while pairs of upper pressure rollers, either spring-loaded or set with an interference fit, hold the part firmly against these drive rollers to maintain concentricity.



Six speeds are possible by changing the position of a drive belt on two three-step cone pulleys, with the output going to a worm-gear reduction unit. Drive-roller brackets are mounted so that the output shaft of the reduction unit passes through a gear in the bracket which meshes with a gear on each roller shaft. Thus, a positive, uniform drive is provided, yet drive-roller brackets can be positioned anywhere along the part to be quenched. Limit stops restrict axial movement and their positioning, along with that of the rollers, may be altered to fit various length parts.





A Designer's Guide to

Vitreous Coatings

Service Properties

By Robert L. Stedfeld
Associate Editor, Machine Design

SELECTION of a porcelain enamel for specific service involves not only knowledge of the types of coatings available, but also knowledge of the properties of these enamel coatings. In a previous article (December, Page 165) types of enamels were outlined. This article presents a summary of service properties, grouped and classified for simplified selection of an enamel.

As previously mentioned, vitreous coatings provide excellent protection in three general service areas: They resist acids, alkalies, chemicals and corrosion; they are suitable for high temperatures and for protection against high-temperature oxidation; and their glass-like properties make them ideal for resisting abrasion and wear. A fourth, nonfunctional area of interest is the possible appearance value of porcelain enamel. Required service properties can often be obtained in a coating that is both attractive and appealing.

Determination of the service properties of porcelain enamels has been a constant area of research within the enameling industry, but because of the multitude of enamels available, progress in systematic classification of properties has not been rapid. Adoption by ASTM of several standard tests for porcelain enamels within the last year or two should help to promulgate use of standard tests for evaluating properties of porcelain enamels.

Present test methods shown in TABLE 1, are designed for comparison of one porcelain enamel against another, rather than for evaluation of enamels in terms of absolute criteria. Intended primarily for standard cover-coat enamels, test methods are adequate for the usual range of conditions but cannot always be extended to cover extreme cases, such as acid-resistance of glass linings.

Optical Properties

Although appearance of the surface is often not the essential consideration in porcelain-enameled components, in most present applications sales appeal and attractive appearance have been prime objectives. Consequently, much work has been devoted to extending range of color, reflectance and other visual factors.

Opacity and Reflectance: Strictly speaking, opacity and reflectance are not the same thing, since opacity is a measure of hiding power and reflectance of ab-

sorption of incident light rays. But for white glossy enamels, reflectance is the practical, commonly used measure of opacity.

Reflectance of 74 to 78 per cent is the usual practical standard for titanium enamels, although reflectance of 80 to 85 per cent is entirely feasible at normal application weights of 20 to 30 grams per square foot. Similarly, zirconium-opacified enamels are usually applied to attain a reflectance of about 75 per cent; acid-resisting grades of antimony enamels normally run 60 to 68 per cent; and nonacid-resisting grades of the same enamel, 73 to 75 per cent.

Titanium enamels are more opaque than either zirconium or antimony enamels. Reflectance of 74 to 78 per cent can be attained with titanium-opacified enamels at application weights of 18 to 20 grams per square foot, whereas 40 to 45 grams per square foot would be required for zirconium enamels and 45 to 55

Table 1—Tests for Service Properties of Porcelain Enamels

Reflectance

Purpose: To determine diffuse reflectance and reflectivity of white porcelain-enamels.

Results: Determination of *reflectance*—fraction of light falling on a porcelain-enameled surface that is diffusely reflected, expressed as a decimal fraction or per cent at a thickness or weight per unit area; and *reflectivity*—reflectance at a thickness so great that additional thickness does not change the reflectance, expressed as a decimal fraction.

Method: Specimens are illuminated at an angle of 45 degrees to the surface in a reflectometer. They are viewed at an angle of 90 degrees and compared with the ideal completely reflecting and perfectly diffusing surface (freshly smoked surface of magnesium oxide in actual practice).

Test: T-13 (PEI); CS144-47 (EUMC).

Acid Resistance

Purpose: To determine acid resistance at room temperature of porcelain enamels for all articles except cooking utensils, and chemical and hospital ware.

Results: Classification of specimens into five classes, designated as Class AA, A, B, C or D.

Method: Three tests are possible—a commercial test, an umpire test and a research test, with increasing degrees of accuracy. The commercial test consists of five steps:

1. *Visual examination.* Specimen is treated with 10 per cent citric acid solution, both at 80±10 F for 15 minutes. Visual examination determines any change in appearance.

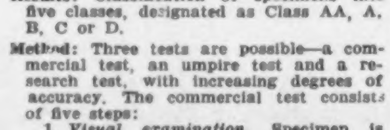
2. *Dry-rubbing test.* Treated and untreated areas are marked with lines made with No. 1 lead pencil, then rubbed with a dry cloth. More tenacious retention of lines in the untreated area indicates failure of the test.

3. *Wet-rubbing test.* Same as dry rubbing test except cloth is wet with water and wrung out.

4. *Blurring-highlight test.* Image of a small light source, such as a frosted bulb, is observed in the treated area. Definite blurring as image passes across boundary from the untreated to the treated area indicates failure.

5. *Disappearing-highlight test.* Same as (4) except that image disappears in treated area.

Enamels are then classified as follows:



Test: C282-51T (ASTM); T-7 (PEI)

Boiling Water Resistance

Purpose: To determine solubility of an enamel in boiling water.

Results: Percentage of initial specular gloss retained after the test is an index of the *solubility*.

Method: Initial gloss is measured on a gloss-measuring instrument on two 3¼ by 4-inch specimens which are then subjected to the action of continuously boiling distilled water at a pH from 5.5 to 7.0 which is continually changed for 7 days. After this period, gloss is again measured.

Test: CS115-44 (EUMC).

Boiling Acid Resistance

Purpose: To determine the resistance of porcelain enamels to attack by boiling acids.

Results: Amount of coating dissolved constitutes the *weight loss*, grams per square inch.

Method: Three 3¼-inch diameter test specimens are weighed, and placed in a boiling solution of 6.4 per cent citric acid for 2½ hours. They are then dried and reweighed to determine the total loss in weight over the area attacked by the acid.

Test: C283-51T (ASTM); CS100-47 (EUMC).

Surface Abrasion Resistance (PEI)

Purpose: To determine the resistance of the surface layer of porcelain enamel to abrasion.

Results: The specular gloss reading after abrasion, divided by the initial reading and multiplied by 100, is taken as the *abrasion index* of the coating.

Method: After measuring initial specular gloss with a reflectometer, six specimens are abraded by oscillating the specimens horizontally under a vertical rubber cylinder containing alloy steel balls, standard feldspar abrasive and water. After approximately 1465 cycles, the specimen is cleaned and specular gloss is again checked.

Test: T-2 (PEI).

Surface Abrasion Resistance (Taber)

Purpose: To determine the resistance of porcelain enamel to harsh abrasion.

Results: Results are indicated as a *weight loss*, milligrams, for a number of cycles under a specified load using a specified abrasive wheel.

Method: Two abrasive wheels revolve on weighted arms through frictional contact with a specimen mounted on a motor-driven turntable. Angle of each wheel is set to produce two crisscrossing arcs on the specimen. In most cases a coarse-grit wheel is used.

Test: Taber.

Gouging Resistance

Purpose: To determine the resistance of the underlying enamel structure to crushing.

Results: The load which produces gouging of 50 per cent of the path of a rolling ball is regarded as the *gouging resistance*, pounds.

Method: A 3/32-inch diameter stainless-steel ball is rolled over the porcelain-enameled test piece under increasing loads. As load is increased, a point is reached at which the ball penetrates the enamel and gouges it out over a small percentage of the total path. As weight is increased, the amount of gouging increases. Gouging of 50 per cent of the path is the end-point.

Test: T-1 (PEI).

Impact Strength (PEI)

Purpose: To determine resistance to mechanical impact of porcelain-enameled test specimens.

Results: Energy of the blow required to produce failure is taken as the *impact resistance*, ft-lb.

Method: A cylindrical specimen having a standard thickness of porcelain enamel is struck by a 1-inch diameter hardened steel ball which forms the "nose" on the ¾-pound head of a 9-inch long pendulum. Blows are struck at increments of 0.0125-ft-lb of energy until the first visible fracture (crack or chip) occurs.

Test: T-6 (PEI)

Impact Strength (ASTM)

Purpose: To determine resistance to mechanical impact of porcelain-enameled articles.

Results: Height from which the test ball must be dropped to produce failure is the *impact resistance*, inches.

Method: The article to be tested is clamped and a guide tube to guide the test ball is located above the specimen. A cold-worked duraluminum ball, ⅞-inch in diameter and weighing 10 grams, is dropped from increasing heights until the first chip visible at a distance of 18 inches occurs.

Test: C284-51T (ASTM); CS100-47 (EUMC)

Torsion Resistance

Purpose: To determine resistance to torsional deflection of a porcelain-enameled test specimen. The test involves both twisting and, to some extent, bending.

Results: The angle of twist producing failure of the coating is taken as the *torsion resistance*, degrees.

Method: A metal blank is bent to form a 1-inch angle, 8 inches long, and is porcelain-enameled to a standard coating thickness. It is then clamped rigidly along one leg of the angle. The other long edge is clamped to a lever arm, which is loaded at the free end by filling a can with water at a specified rate until the coating first chips at the apex of the specimen. Deflection angle is read by noting position of the free end of the lever arm against a calibrated scale.

Test: T-5 (PEI).

ASTM—American Society for Testing Materials; PEI—Porcelain Enamel Institute; EUMC—Enameled Utensil Manufacturers' Council, Commodity Standards Div., U. S. Dept. of Commerce; Taber—Taber Instrument Corp.

grams for antimony enamels.

As high as 90 per cent reflectance can be attained with titanium enamels. Excessive thickness of enamel would be required to accomplish this reflectance with the other two types. In comparison to these figures, enamels for aluminum have a maximum reflectance of 77 per cent, which is reached at a total thickness of 0.015-inch.

Surface Finish: Porcelain enamels are normally available in all degrees of luster or surface texture, from a high gloss to a full matte. Eggshell, pebble and similar "rough" finishes can be attained, and even "nonslip" surfaces have been accomplished by sprinkling a standard enamel with grains of refractory material.

Color: As previously mentioned, all degrees of color can be attained in enamels. Titanium enamels at the present time are unsuitable for anything but white or light pastel colors produced by tinting with color oxide. Within the range of cream-white to blue-white, however, color matching can be extremely close between antimony, zirconium and antimony-base enamels.

Strong, pure colors are possible in all colors of the spectrum, as well as browns and other "mixed" colors. Chroma (purity or dullness) of the colors is dependent on composition of the enamel. Lead in an enamel, for example, tends to brighten most colors with the exception of reds, oranges and cadmium yellows. Blue appears brighter if there is zinc oxide in the enamel, but certain shades of green are deadened under exactly the same conditions.

Colors in acid-resistant enamels are noticeably more resistant to fading by weather conditions than in nonacid-resisting types. Class AA and A enamels (TABLE 1) measured in one seven-year test showed no noticeable fading. Class B enamels were slightly faded—but not objectionably. Noticeable color changes, however, occurred on Class C and D enamels.

Acid and Corrosion Resistance

Enamels can be formulated for all degrees of acid resistance. Limitations exist on degree of alkali resistance, but enamels can be formulated to withstand mild alkalies at temperatures up to boiling.

Acid Resistance: The test for acid resistance outlined in TABLE 1 gives a range entirely satisfactory for determining resistance to food and other mild acids in "normal" environments. Standard white opaque enamels can be formulated in all grades from Class D to AA, although only Class A enamels or above are considered "acid-resisting." Titanium-opacified enamels and the acid-resistant grades of antimony enamels fall into this range quite easily, since they can both be formulated in Classes B to AA—although the antimony types can have higher acid resistance than titanium enamels. Nonacid-resisting antimony enamels and zirconium enamels are both rated Class D.

Relative resistance of standard enamels to boiling acids is shown by the following table, in which the enamels have been tested with 10 per cent citric acid solution instead of the 6.4 per cent solution mentioned in the boiling acid test given in TABLE 1; weight loss is expressed in grams per square inch for each type of enamel.

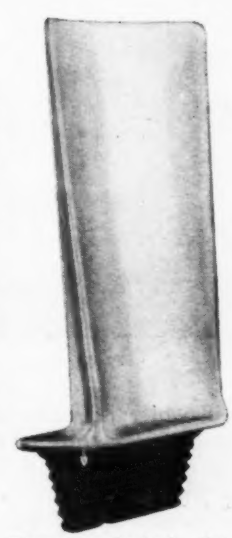
Antimony (acid-resistant)	0.0008
Titanium	0.0015
Zirconium	0.0130-0.3400

Acid resistance of ceramic coatings (Solaramic) is approximately equivalent, with a 0.005-gram per square inch loss when tested with 6 per cent boiling citric acid for 2½ hours.

Extremely high acid resistance is given by "glass linings" and similar types of enamels at room temperatures. Offering protection against all acids except hydrofluoric and hot concentrated phosphoric,

Table 2—Boiling Acid Action on Glass-Lining Enamels

Acid	Concentration (% by weight)	Penetration (0.0001-in. per year)	Acid	Concentration (% by weight)	Penetration (0.0001-in. per year)
Hydrochloric ..	5	5.13	Nitric ..	5	2.83
	10	3.06		20	12.02
	20	0.33		40	0.08
Sulphuric	5	5.83		69	0.86
	20	14.78	Acetic ...	5	2.60
	40	1.41		40	0.93
	80	None		99.5	0.41



Intense heat plus high stresses cause fast deterioration of jet-engine buckets. One of the newest attempts at solving the problem is the combination of molybdenum and a ceramic coating. Molybdenum, with a melting point of 4750 F, has high-temperature strength, while the ceramic coating protects against high-temperature oxidation.

Photo, courtesy Thompson Products Inc.

such enamels have a very slow rate of attack even at boiling temperatures, as shown in TABLE 2.[1]

Adequate resistance is possible at even higher temperatures with glassed steel. Sulphuric acid in concentrations above 10 to 15 per cent can be handled at temperatures up to 450 F, and temperatures up to 300 F are possible for all concentrations of hydrochloric. Even chlorosulfonic acid, one of the toughest of corrosives, has little effect at normal temperatures.

Alkali Resistance: In general, porcelain enamels are not resistant to hot alkalies. "Standard" commercial formulations give protection against mildly alkaline solutions, such as soaps and water softeners, of pH 7 to 10 at temperatures up to about 140 F. Often, commercial ground coats are used as the final coating to give alkali resistance; however, other formulations as shown in TABLE 3 seem to be more satisfactory.

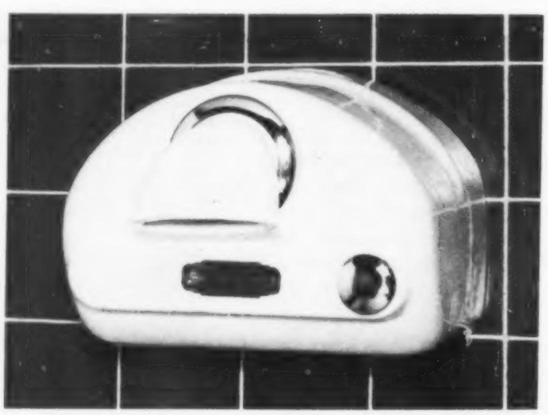
The data in TABLE 3 are for samples immersed for 2½ hours in a boiling solution of 5 per cent sodium pyrophosphate in a test[2] similar to that described for boiling acids in TABLE 1—a procedure which has been suggested as a method of testing for alkali resistance. It is interesting to note that a ceramic coating has excellent alkali resistance, although it is not used for this type of service.

Table 3—Effect of Boiling Alkaline Solution on Porcelain-Enamel*

Enamel	Weight Loss (gm per sq in.)	Visual Appearance
Alkali-resistant second coat..	0.001	Slight etch
Alkali-resistant overspray ...	0.002–0.003	Slight etch
Alkali-Resistant ground coat.	0.003	Etched more than second coat
Commercial black edge overspray	0.006–0.021	Severe etch—matte finish
Commercial ground coat.....	0.008	Etched—matte finish
Solaramic ceramic coating...	0.0007	†

* 5 per cent sodium pyrophosphate solution for 2½ hours.
† Information not available.

[1] *References are tabulated at end of article.*



Photo, courtesy Chicago Hardware Foundry Co.

Dirt, water corrosion and low heat must be resisted by the case on this hand dryer. The housing, a casting coated with white porcelain enamel by the dry process, must be able to withstand repeated washing and maintain a neat, sanitary appearance.

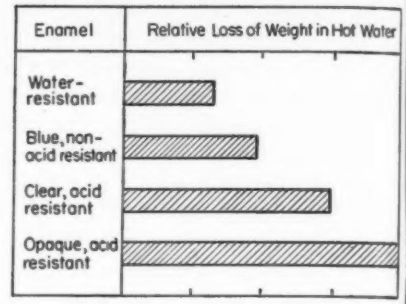


Fig. 1—Relative resistance of porcelain enamels to corrosive action of hot water

Again, in the glass-linings field, formulations have been developed which will withstand hot alkalies. The latest one is reported to withstand an alkalinity of pH 12 at 212 F with no reduction in acid resistance.

Hot Water Resistance: Resistance of porcelain enamels to hot water cannot be gaged by acid resistance of an enamel. Hot water, particularly under pressure, is extremely corrosive. Acid-resistant enamels are not hot-water resistant, but all hot-water resistant enamels have some appreciable degree of acid resistance.

The difference lies in the method of attack. Acids attack the chemical composition of the enamel, whereas hot water tends to dissolve the frit. The effort, therefore, in formulating an enamel resistant to hot water is to decrease solubility. Some idea of relative resistance to hot water can be gained from *Fig.* 1.

Higher temperatures increase severity of attack, and steam is particularly corrosive. The few failures reported, however, for porcelain-enameled or glass-lined hot-water tanks have not been the result of failure of the enamel to withstand attack by hot water, but have been localized corrosion caused either by a defect in the enamel or by a break in the coating.

Table 4—Thickness of Enamel Coatings

Type of Enamel	Thickness (in.) Minimum	Average	Maximum*
Sheet-steel ground coat......	0.002	0.003-0.005	....
Titanium enamel†	0.0015	0.003-0.004	0.006
Zirconium enamel†	.....	0.006-0.008	0.015
Antimony enamel†	.....	0.006-0.009	0.025
Aluminum enamel ground coat	.....	0.002-0.004	....
Aluminum enamel cover coat†	.....	0.004-0.009	0.012
Dry-process enamel	.....	0.015-0.040	0.050
Ceramic coating	0.0005	0.001-0.003	....
Glass lining (dry process) ...	.....	0.025 up	....

* Represents normal limit, or point of maximum reflectance; extra-heavy coatings can be built up. † Not including ground coat.

Porcelain enamels compared with galvanized coatings, for example, have excellent hot-water resistance but must be continuous, without breaks or faults.

Weather Resistance: Extensive tests[3, 4] have been conducted to determine effect of weather on porcelain

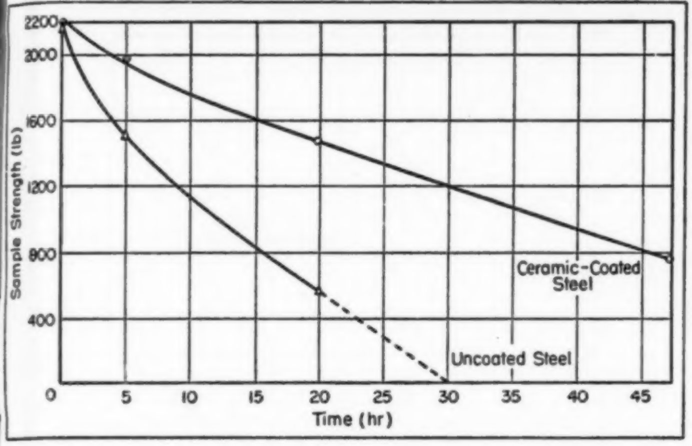


Fig. 2—Effect of extended heating at 200 F on the strength of uncoated and ceramic-coated type 347 stainless steel

enamels. The general conclusion from these tests is that all enamels adequately protect the underlying iron from rusting. Some enamels, however, proved better than others in resisting the combined effects of heat, cold, rain, sunlight, gases and smoke.

Acid-resistant enamels are, as a whole, distinctly more durable than nonacid-resisting. Formation of pits in nonacid-resisting enamels, however, can be prevented by a thin overglaze of clear, acid-resisting enamel. High concentrations of combustion or other corrosive-acid gases seem to have a pronounced effect on weathering.

Full matte-finish enamels do not seem to be as satisfactory for outdoor service. Although resistant to weathering, such enamels are subject to stains from foreign substances in the atmosphere, whose severity is increased with increase in degree of matteness. Full matte enamels appear to be unsuitable where appearance is important because of fading. (Also see *Color.*)

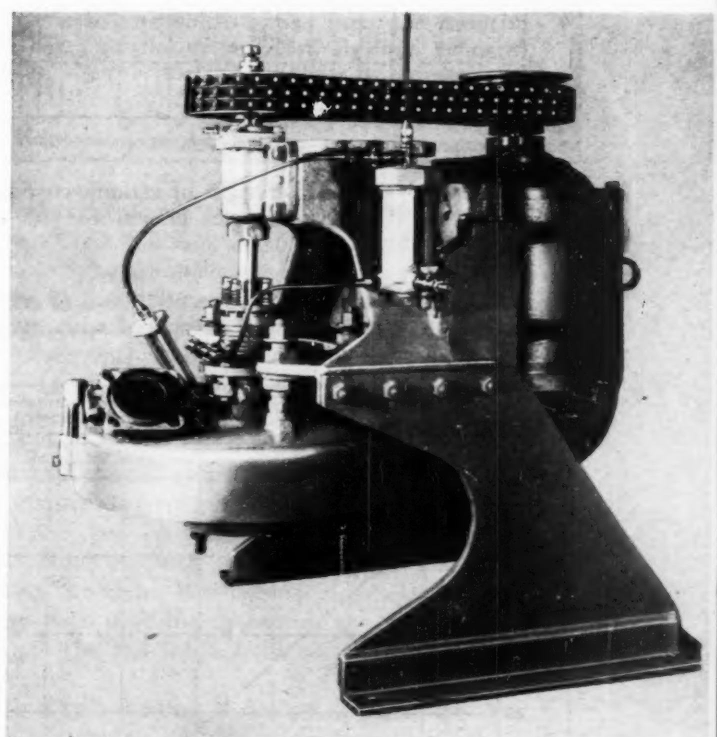
Electrolytic Corrosion: Since porcelain enamels are basically glasses, there is no possibility of electrolytic corrosion unless there is a defect in the enamel or the coating is damaged. Any corrosion resulting from such a defect will usually be localized in nature, with surrounding areas remaining protected.

Thermal Properties

Of importance in any consideration of temperature resistance of porcelain enamels are two main factors: heat resistance and resistance to thermal shock. Heat resistance is a relative term; what is usually meant with ceramic coatings is the ability of the coating to protect the underlying metal (usually ferrous) from oxidation. Oxidation at high temperatures is greatly accelerated, often causing complete disintegration within a short time. Carbon absorption, attack by lead-bromide vapors and corrosive attack may also add to the problem in engine exhaust systems. Each application poses special problems—but the prime problem is that of high-temperature oxidation.

Temperature Resistance: "Standard" porcelain enamels can be used up to 800 to 1000 F—a range much higher than conventional organic finishes. Below this range the enamel coating is unaffected. From 1000 to about 1200 F the enamel may be slightly damaged, depending on softening temperature, but generally the underlying metal will be well protected.

For the temperature range from 1200 to 2000 F or higher, ceramic coatings offer excellent protection to ferrous-metal parts. The exact high limit usually depends upon expected service life and the exact application. With present ceramic coatings, extended service life at temperatures in the range of 1500 to 1700 F seems to be entirely feasible. Localized temperatures may run as high as 2000 to 2200 F. With some heat dissipation, long life may be expected for parts subjected to these higher temperatures over their entire surface. Ceramic-coated molybdenum has been subjected to temperatures of 2800 to 3000 F with satisfac-



Photo, courtesy Pfaudler Co.

High chemical resistance, as well as capacity to handle high temperatures and pressures, is provided by the glass lining of this low-speed centrifugal pump. Both the impeller and pump housing are cast iron with a glass coating

tory life for some uses.

For such applications as jet or piston engine parts for aircraft, the requirement may be only that the part have a service life measurable in hours instead of months or years. In these applications, ceramic coatings can usually improve the performance of the base alloy. *Fig.* 2, for example shows the effect of heating samples of type 347 stainless steel—not an extremely high-temperature alloy—at 2000 F under tension. Although loss in strength is great, the ceramic-coated specimen shows a decided advantage over the uncoated specimen for this rather drastic test.[5]

Fatigue strength at high temperatures is also improved when ceramic coatings are used. Some indication of the increase can be gained from *Fig.* 3, showing the difference in fatigue strength and fatigue life between coated and uncoated specimens of type 347 stainless heated for 10 hours at 1800 F.

Nonstructural applications, however, represent the largest area of usefulness for ceramic coatings. In these applications protection from oxidation is the

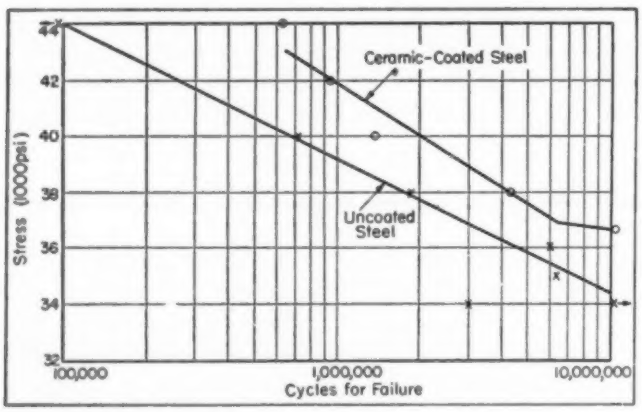
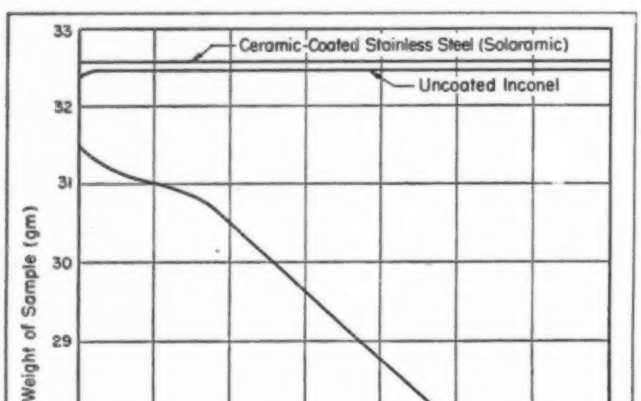


Fig. 3—Above—Fatigue strength of ceramic-coated and uncoated type 347 stainless stel heated for 10 hours at 1800 F

Fig. 4—Below—Rate of oxidation at 1700 F of ceramic-coated type 321 stainless steel compared with uncoated steel and Inconel



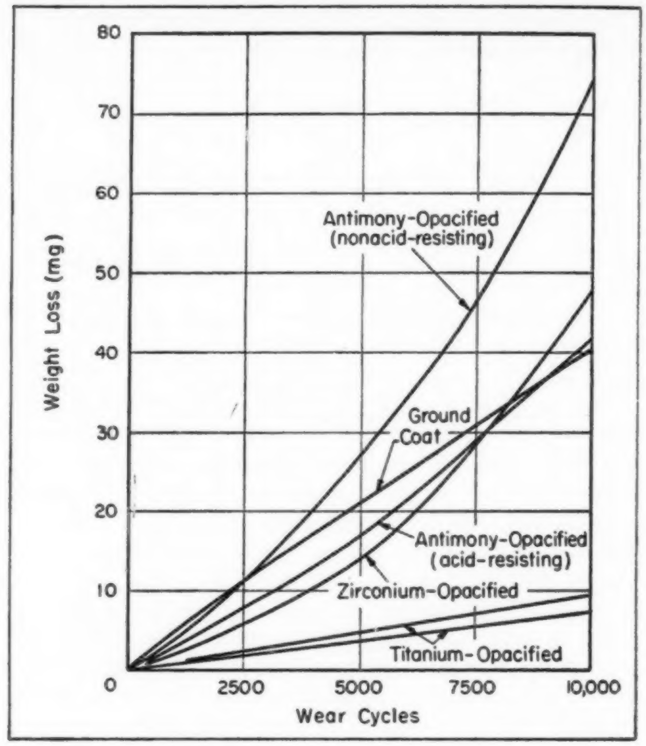


Fig. 5—Abrasion resistance of standard enamels tested in a Taber Abraser

paramount criterion, and not the maintenance of structural strength. Thus *Fig.* 4 shows that type 321 stainless steel with a ceramic coating suffers no oxidation when continuously heated to 1700 F.

Thermal Shock: Resistance to rapid changes in temperature must be high, especially for ceramic coatings which must withstand temperature fluctuations of a large magnitude within a very short time.

Standard porcelain enamels are able to withstand a temperature drop of 200 F without any damage, and some can withstand heating to 350 F, then immediate dipping in ice water over a number of cycles without damage. In other tests, samples enameled with one ground coat and two cover coats were heated to 300 F and quenched in water at room temperature. The process was repeated at 100 F intervals up to 1600 F. Microscopic hairlines, invisible to the naked eye, began to form at 400 F, but did not become visible until the samples were heated to 1600 F.

Thermal shock properties of ceramic coatings are much better, and more drastic tests are used. Solaramic coatings, for example, are:

1. Heated for 45 minutes to temperatures of 1850 F, 1950 F, 2000 F and 2100 F, followed by a 15-minute air quench. This test is repeated 9 times and the part is then soaked at temperature for 15 hours.
2. Coated parts are subjected to the same test at a temperature 100-200 F higher than maximum expected operating temperature. Testing is continued for 200 hours.
3. Parts are heated to 1700 F and water quenched through 10 cycles.

In addition, ceramic-coated parts are tested for hot-

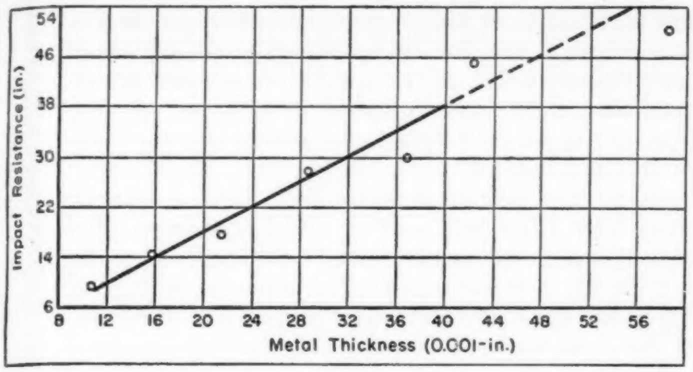
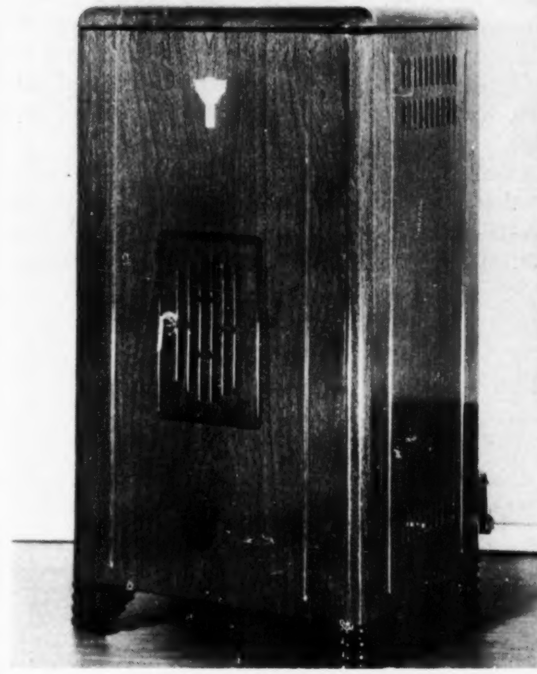


Fig. 6—Effect of metal thickness on impact strength of 0.013-inch thick porcelain-enamel coatings



Photo, courtesy Vitreous Steel Products Co.

Unusual finishes can be attained with porcelain enamels, as evidenced by the wood-grain finish on this enamel-coated space heater. Applied by a special graining roll, the finish is "printed" on the metal part which is then fired. In addition to being unaffected by relatively high temperatures, the porcelain enamel must prevent rust.

spot resistance and "flame" resistance, both tests of thermal-shock characteristics. These tests are:

1. Spot heating with an oxyacetylene torch to 1700-1800 F and water quenching through 10 cycles.
2. Heating a rolled, spot-welded and ceramic-coated specimen for 15 minutes in the hot blast of a gas-air burner to 1700-1900 F, then air cooling for 15 minutes. Test is conducted over a 16-hour period.

Extremely important in all parts subjected to thermal shock service is thinness of the coating and close matching of coefficient of thermal expansion of the enamel with that of the base metal. Most ceramic coatings are formulated for application in the thinnest possible layer. Sometimes an effort is made to select a frit with a high coefficient of expansion to resist thermal shock.

Heat Flow: A third factor, which has an effect on both thermal shock resistance and fatigue strength at high temperatures, is heat flow through the coating. One of the main problems in using ceramic coatings has been to reduce hot spots, thus increasing fatigue strength by eliminating stress raisers caused by strong temperature gradients. A thin coating has proved to be beneficial. In addition, emissivity of the coating can be adjusted to provide a heat-reflective coating on the inside of hot parts and a coating which radiates heat on the outside.

Mechanical Properties

Porcelain enamels are roughly comparable to organic coatings in resistance to mechanical damage. Although glass-like, enamels are strongly bonded to a steel backing, and are thus "reinforced." Design of the component and selection of the base metal play an important part in developing the most favorable properties of the enamel.

In general, flexibility of an enamel, and consequent resistance to bending, warping and twisting, depend on thickness of the coating. Thin coatings withstand a considerable degree of distortion without fracturing; however, thick coatings must have rigid support. Impact resistance is moderate, but here again proper design for the service requirements involved will help to develop adequate impact resistance. Being hard and glossy, vitreous coatings offer excellent protection against abrasion and wear.

Thickness: Thickness of a porcelain coating, dependent on the enamel chosen and method of processing, can be controlled within limits. Dry-process enamels are thicker per coat than wet-process enamels, and can be built up with multiple coats to almost any thickness desired. Wet-process enamels, too, can be built up with multiple coats, but the range is usually limited to the thickness attainable with three or four coats.

Often a thin coating is desired to minimize the possibility of damage by impact or thermal shock; however, a thick coating may likewise be an advantage for added protection against acid or chemical attack. Minimum thickness is usually limited by the type of enamel and the number of coats of the specific type which must be applied. Ranges of enamel thickness are given in TABLE 4.

Some unusually thin coatings have been used. Ceramic coatings for example are sometimes applied as thin as 0.0005-inch. Even standard two-coat white

opaque enamels have been used in total thicknesses as little as 0.005-inch—an application involving the automatic spraying of an 0.002-inch ground coat and 0.003-inch cover coat on paper-thin 0.001-inch thick steel for bonding on hard wallboard. Such thin coatings can usually be bent, punched, sawed and otherwise cut without excessive chipping or flaking at the cut edge.

Abrasion and Wear Resistance: Porcelain enamel's good resistance to wear and abrasion is due to a combination of three properties: resistance to gouging or crushing of the underlying enamel structure, high surface hardness (surface abrasion resistance), and high surface gloss.

Table 5—Torsion Resistance at Various Coating Thicknesses

Enamel	Torsion Resistance (deg) at 0.006-in.	at 0.009-in.	at 0.015-in.
Titanium enamel	70	41	32.9
Zirconium enamel	..	32.5	27.0
Antimony enamel	..	31.6	24.0

Table 6—Radius of Bend Producing Enamel Failure

20-gage Metal (0.0375-in.) Enamel Thickness (in.)	Failure Radius (in.)	32-gage Metal (0.0101-in.) Enamel Thickness (in.)	Failure Radius (in.)
0.0023	8.0	0.0015	2.3
0.0033	9.1	0.0030	3.2
0.0040	10.6	0.0063	5.2
0.0085	11.8		

Hardness of porcelain enamels varies from $3\frac{1}{2}$ to 6 on Moh's scale, compared with a Moh's hardness of 2 to 3 for organic finishes. Abrasion-resistant enamels are harder and denser than ordinary types, and have a high gloss. Some are so hard that it is difficult to scratch them with a tool-steel blade, and these grades resist abrasion better than many metallic coatings.

Surface abrasion resistance, as measured by PEI test (TABLE 1), is from 30 to 60. Reproducibility of this test has been found to be low, however, and a more accurate indication of resistance to surface abrasion is given in *Fig.* 5. In this latter test, samples of standard enamels were abraded in a Taber Abraser using two type CS-17-F Calibrase wheels under a 1000-gram load per wheel.[6]

One of the factors which helps porcelain enamels resist wear is its high surface gloss, smoothness and consequent low surface friction. In materials-handling, the "slick" qualities of porcelain enamels, and their low resistance to the flow of viscous or abrasive fluids or solids increases their wear life considerably.

Resistance to Chipping: Resistance to chipping by mechanical action may depend upon many factors, among them flexibility of the enamel, degree of adherence, thickness of the enamel, and design of the part. Each of these may, in turn, depend on process or manufacturing variables. Thus "chipping resistance" becomes a highly inexact term.

Tests for determining chipping resistance of porcelain enamels fall into three classes: (1) impact tests, (2) torsion, bending, twisting or deflection tests and, (3) stretching tests. The third type, requiring extensive laboratory equipment, is not generally used except in research.

IMPACT RESISTANCE: Impact resistance of porcelain-enameled specimens varies greatly with design. A thicker base metal for example, can increase resistance to impact,[7] as shown in *Fig.* 6. Larger radii also seem to aid in resisting impact, *Fig.* 7. A thicker enamel seems to have higher resistance to chipping on radii, as shown by *Fig.* 8, in which blows were struck on the turned edge of an enameled sheet.[8]

BENDING AND DEFLECTION: In the standard PEI test for torsional deflection, porcelain enamel coatings show less resistance to chipping as coating thickness increases, TABLE 5, in direct opposition to the previously mentioned relationship between thickness and

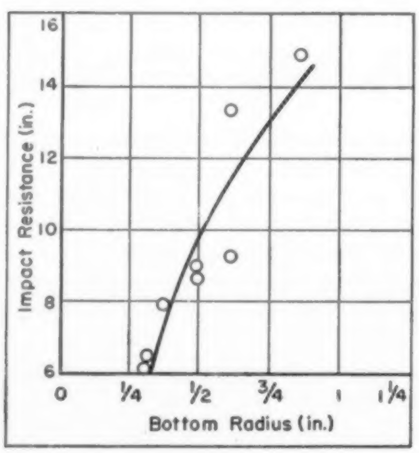


Fig. 7—Left—Effect of increasing bend radius on impact strength of enamel coatings on 28-gage (0.0156-inch) metal

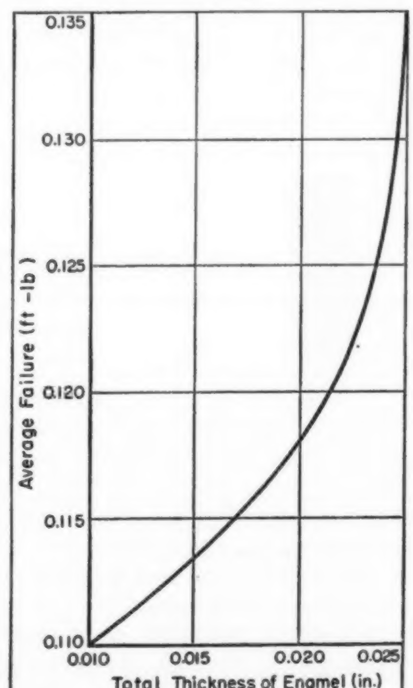


Fig. 8—Right—Impact strength as a function of thickness of the enamel coating

impact resistance. The relationship shown in TABLE 5 has been confirmed by bending tests, TABLE 6, in which an enameled test specimen is bent in diminishing radii until the coating fails.[9] Thus, in bending or twisting, the thinner coating is more desirable.

INCREASING CHIP RESISTANCE: The seeming paradox posed by the relation between enamel thickness, impact tests and bending tests is resolved when we consider the method of making the tests. Impact tests are generally made on a bent edge or cylinder which is inherently stiff. Thus, the enamel coating is not deflected before chipping as much as in the bending tests.

It seems, therefore, when extremely rigid parts are to be enameled, a thicker coating would be more desirable. Since extreme rigidity is impossible in most sheet-metal parts, the opposite direction is usually taken, and effort is made to reduce coating thickness, thus increasing flexibility of the enamel. Other methods, such as increasing radii or stiffness of the component, are design considerations which will be covered in a later section.

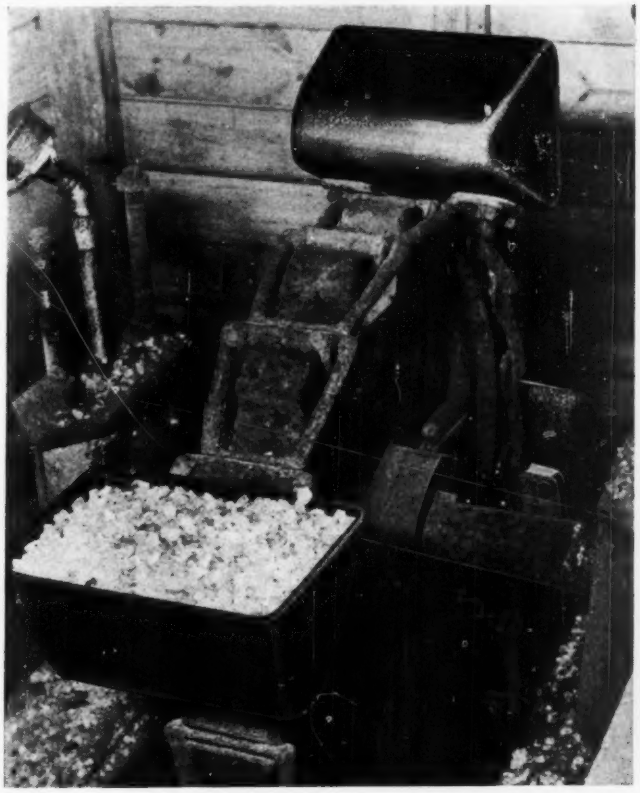
Miscellaneous Basic Properties

Standard porcelain enamels are basically alkaline borosilicate glasses, and have much the same properties. Thus the following properties of standard porcelain enamels bear much resemblance to those of the corresponding glass:

Specific gravity	2.5-3.0
Specific heat (cal/deg C/gram)	0.20-0.25
Compressive strength (psi)	30,000
Tensile strength (psi)	6000-7000
Coefficient of linear expansion (in. per in. per deg F)	$8\text{-}12 \times 10^{-6}$
Thermal conductivity (cal/sq cm/cm/sec/deg C)	0.001-0.003

Summary—Enamel Selection: Selection of an enamel is not simple, since enamels are not available in standard formulations like plastics or steel. Often formulations are varied slightly to adapt the enamel to a particular service condition or to individual peculiarities of a porcelain enameler's processing technique.

Enamel selection, component design, and processing technique must be closely related for effective utilization of a vitreous coating. Both design and processing critically influence selection of a specific enamel, and should be considered at the time of enamel selection. When an enamel is to be selected, therefore, minimum information at hand should include (1) a resume of anticipated service conditions and required enamel properties, (2) the preliminary or tentative design, so that revision can be made if processing or enamel selection necessitate a change and (3) a thorough knowledge of the processing capacity of the enameler.



Photo, courtesy Porcelain Enamel Institute

Extreme conditions of abrasion and corrosion are encountered by these elevator buckets in handling rock salt. A porcelain-enamel coating has proved to be very effective in meeting the rugged service requirements, providing long life and requiring infrequent replacement.

The main considerations in design for enameling, selection of a base metal, and processing will be covered in a third and final article of this series to be published next month.

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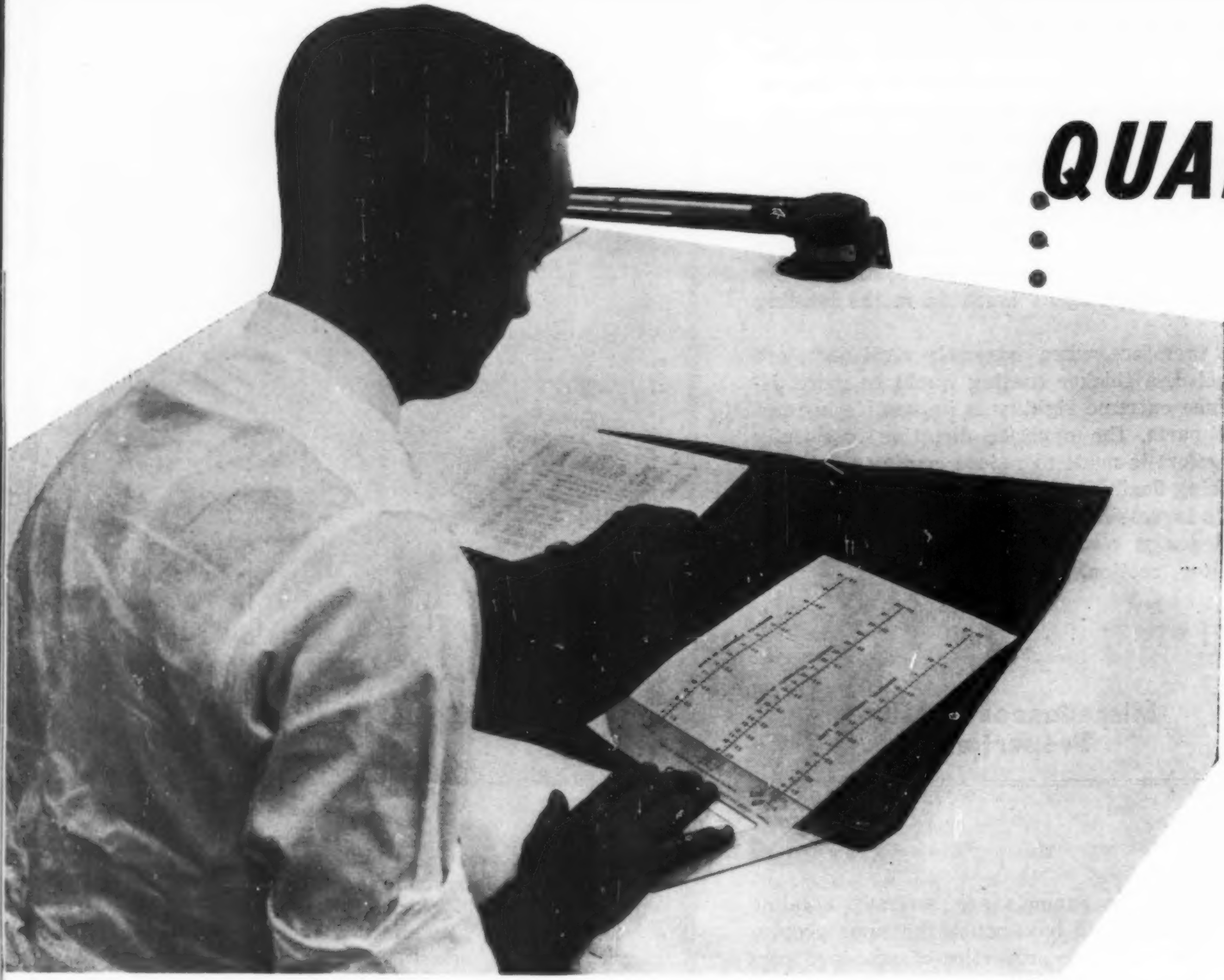

QUALITY C

Part 7—Determining Practical Tolerances

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Roth & Strong Inc.
Boston, Mass.

A TOLERANCE limit marks a boundary. It should have this importance. By keeping parts within drawing limits, assemblies should be satisfactory. However, designers must give consideration to the question, "What if just a few characteristics of the parts are somewhat beyond tolerance? Will the assemblies go together and function properly?"

Earlier in this series it was pointed out that designers seldom know the real limits, if rather sharp dividing lines even exist. Extremely conservative persons caused C. G. Darwin to state, after making a survey on how tolerances were assigned,* "To exaggerate the picture which I got as a result of my inquiries, I concluded that in designing a new machine the chief engineer drew it freehand with dimensions to the nearest inch and sent it to the draftsman to work out the detail to the nearest thousandth, who then gave it to his junior assistant to mark in the tolerances. Instructions were certainly always given that tolerances should be as easy as possible, but only lip service was done to them and the junior assistant, anxious not to get himself into trouble, would, as a general

* C. G. Darwin—"Statistical Control of Production", *Nature*, Vol. 149, pages 573-575, May 23, 1942.

Fig. 41—Above—A table of natural tolerances from shop control charts and a nomogram, *Fig.* 43, permit the designer to determine realistically how tolerances can be expected to actually combine

CONTROL METHODS Their Use in Design

Not only can SQC help establish practical part tolerances but also can make possible improved assembly tolerances at reduced cost in production

rule, think of the smallest number he knew and then halve it." Exaggerated, yes, but emphatic!

The goal in design, of course, is to achieve the best balance among the interlocking factors of satisfactory performance, cost, a large tolerance, and available production equipment, *Fig.* 41. A so-called practical designer will draw on a wealth of experience, handbooks or consultations with shop people and his tolerances may begin to look reasonable—closer to that desired balance point. In considering the three kinds of limits—natural, practical and drawing tolerances—proper design requires that the last two be equal and at least one-third larger than the natural tolerance.

How close can the practical designer come to this goal? Normally his sources of information cannot bring him nearly as close as can factual control chart data. Not only does the written record help immeasurably, but most important is that a process run with a chart will give results that vary less than those usually obtained without this guide.

Natural Tolerances From Control Charts: With a process operating under statistical control, the six standard-deviation spread of individual pieces is, as previously mentioned, termed the natural tolerance of the process. A process is properly considered as in control only when all the average and range points from a run of 10 to 25 or more consecutive samples lie within the control limits.

On an average and range chart the natural toler-

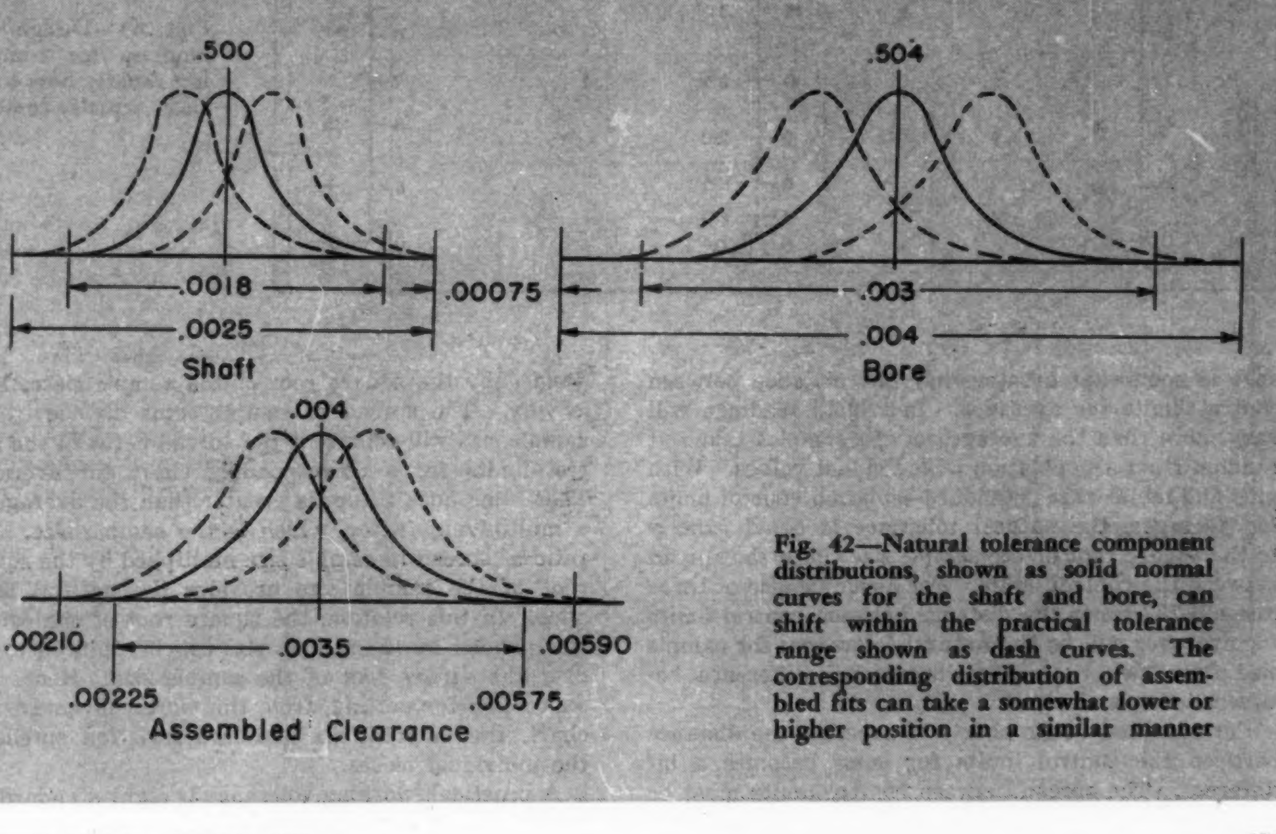


Fig. 42—Natural tolerance component distributions, shown as solid normal curves for the shaft and bore, can shift within the practical tolerance range shown as dash curves. The corresponding distribution of assembled fits can take a somewhat lower or higher position in a similar manner

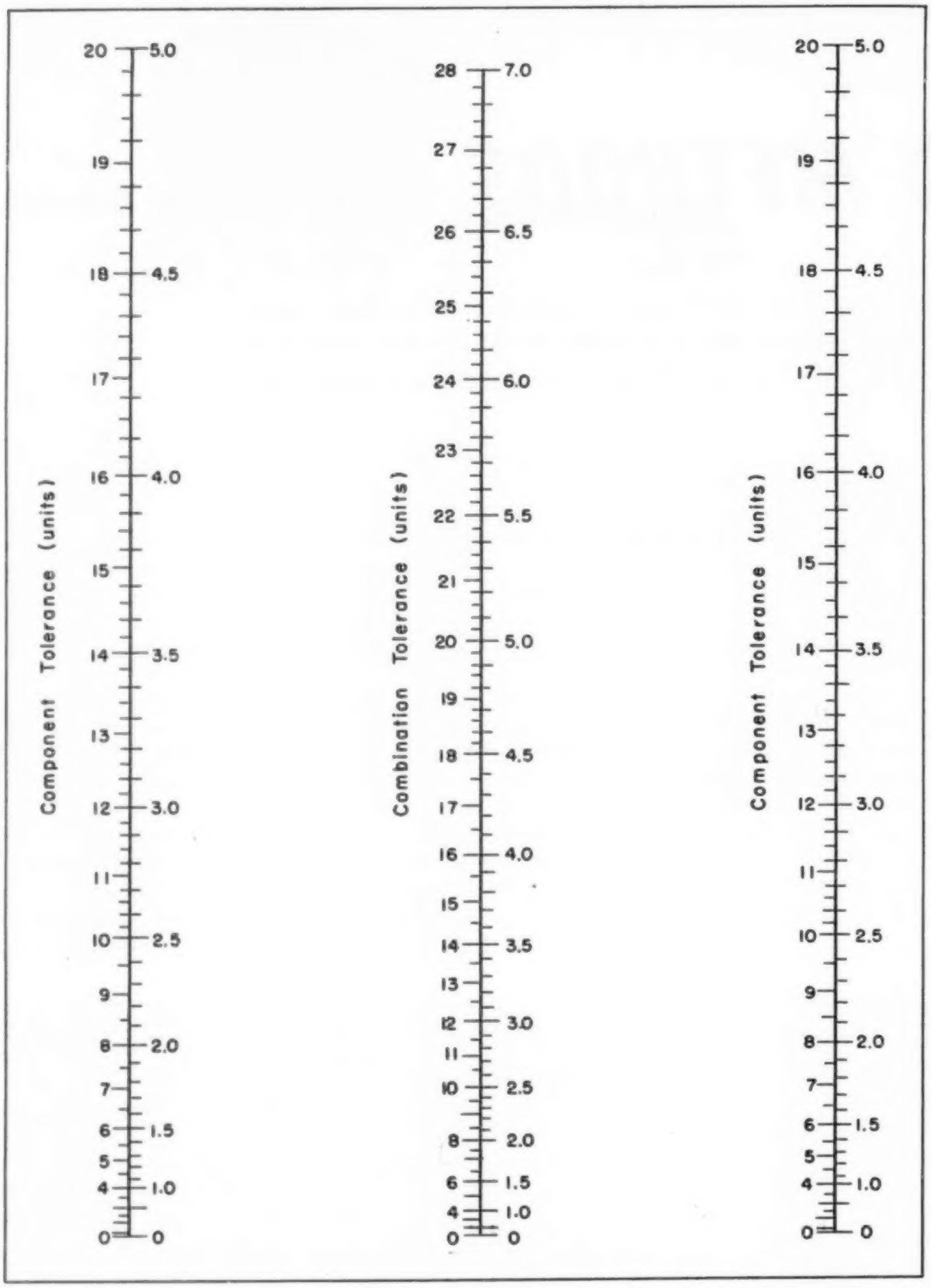


Fig. 43—Design nomogram for computing rapidly how tolerances actually combine

ance is somewhat greater than the distance between control limits for averages. Individual readings will vary more than the averages of the samples taken at random from a population of individual values. With plus and minus three standard-deviation control limits for averages, the natural tolerance is equal exactly to the square root of the sample size times the spread between the control limits. For a sample size of three items, 1.732 times the distance between control limits for averages will be the natural tolerance; for sample size four, two times the dimensional difference between limits is used; etc.

For sum and range charts, converting the distance between the control limits for sums becomes a bit different. The spread between control limits must be *divided* by the square root of the sample size. Here is why. The spread of sample sums divided by the sample size will represent the spread between the control limits for a corresponding chart for averages. That is because a sum is greater than the average by a multiplying factor, which is the sample size. The ratio of spread to sample size multiplied by the square root of the sample size provides the natural tolerance. In this relation, the square root of the sample size divided by the sample size can be reduced to one over the square root of the sample size. Hence, the expression for getting, from the more convenient sum chart, the inherent six standard-deviation spread of the individual pieces.

A practical, working tolerance is eight standard de-

viations, or one-third more than the natural tolerance. Experience has shown this increase allows for repeated settings of the same job, for the use of new tools or dies that are not exactly the same.

Lists of natural tolerances found from shop control charts should be available to designers. These can then be increased by one-third to obtain practical tolerance data. For each problematical tolerance of a new design, the best approach is to consider the most similar fabricating situation by referring to the drawings of parts so listed, TABLE 5, Part 6. Then, the assignment of such practical tolerances to the new design should be considered as a first approximation toward reaching that best-balance goal desired. At this stage it is usually handy to confirm the potential availability of the particular pieces of production equipment listed for running the new job.

How Tolerances Combine: The factors of satisfactory performance and cost will be affected markedly by how tolerances actually combine. This is in contrast to how they could accumulate theoretically.

The correct use of a control chart permits the free action of only variables truly inherent to the process. The variable effects are at the particular values they may have at the moment purely by chance. These conditions give a distribution of the measured values versus their frequency of occurrence that approaches the normal curve as a limit.

The tails of a normal curve contain rather few of all the items produced. The chance selection of two or more parts from the storeroom will seldom result in an assembled fit as severe as one might calculate from using the extremes of the natural tolerance bands. The probability of getting one part of a size 3σ higher than the average of those parts is one to two per thousand. That of getting each of two mating parts that far on one side from their group averages is the product of the separate probabilities, one to four per million, and so forth. Picking many such assemblies of a pair or more of parts will result in an accumulation of final fits that plot also as a normal curve. It will picture the relative probability of getting the fits shown on the horizontal scale, which is size rather than fit in the case of the distributions of the individual components, *Fig.* 42.

Chance or random assemblies guarantee the width of the distribution of the accumulated tolerance to be less than but related to the width of the sum of the component distributions. Actually, the assembly distribution has a "variance" equal to the sum of the "variances" of the components. The variance is the standard deviation multiplied by itself. So the assembly standard deviation equals the square root of the sum of the squares of the component standard deviations.

Natural tolerances can be expected to add in this favorable, statistical manner whenever each is kept within practical tolerance limits with a control chart. The nomogram of *Fig.* 43 makes the statistical addition of any number of natural tolerances easy for the designer.

Consider the example of the fit of a shaft in a bore illustrated by the diagrams given in *Fig.* 42. The nat-

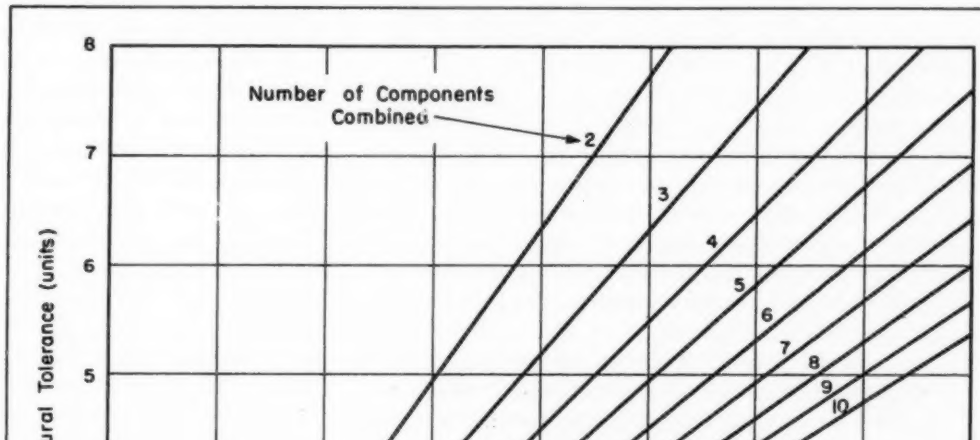


Fig. 44—A substitute chart for the one of *Fig.* 43 for cases when all part tolerances are equal. It is also useful for finding component tolerances starting with a given assembly tolerance when more than two parts are involved

ural tolerance of the shaft is 0.0018-inch. Similarly, the smallest bore that occurs about one to two times per thousand pieces is 0.003-inch less than the largest bore that can be expected to occur at this same rare rate. It is incorrect to compute by straight addition that the maximum spread of the assembled clearance could be 0.0048-inch. Instead use the nomogram and, for this example, read only the magnified scales to the right of each of the vertical lines.

Connect 1.8 on the left line and 3.0 on the right one with a straight edge. The intersection on the center line is read as 3.5. For other needed data, note that the average of the distribution of shaft diameters is 0.500-inch and that the average bore is 0.504-inch. The center of the assembly or fit distribution will be the difference between these two averages, or 0.004-inch clearance. The 3.5 read from the center scale stands for 0.0035-inch total spread of fits equally above and below the 0.004-inch nominal clearance. So the smallest clearance to be expected, about one to two times per thousand, is 0.00225-inch; while the largest occurring at this same rate will be 0.00575.

This nomogram can also be read in reverse order, which more often is the way a designer employs it. Select a certain desired maximum variation for the fit of the assembly. Place this value on the center scale and read off on the side scales for any given position of the straight edge, the amount of tolerance that can be allowed in manufacture for each of the two components.

If either component tolerance runs over five units it will be off the right-hand scale and the three left-hand scales must be used. Five units occurs near the bottom of the left-hand side and the scales run up to 20 units. For readings larger than 20, the right-hand sides of the scales can be used, reading 2.0 as 20, etc. In this respect it is an unlimited nomogram.

In finding the accumulated tolerance of more than two components, the nomogram is also unlimited. With the combination reading of any two of the components used on one of the side scales and the third tolerance in the other side scale, the reading on the center scale is now the statistical sum of the three. This practice can be continued for any number of parts by shifting the center reading to the side before "adding" another tolerance.

For working backwards from an assembly requirement of more than two components, the nomogram must be used by trial and error. Start with the center-scale assembly requirement and read the side scales from a straight edge—one will be the tolerance for one component and the other a combined tolerance for all the remaining components. This combined tolerance can then be entered on the center scale and two more readings on the side scales observed, one being for another component tolerance. The other represents the balance of the combined figures, or the remaining part tolerance if only three pieces are involved.

This trial-and-error method can be helped by an estimate from the chart in *Fig.* 44 which gives the units of component natural tolerance from an assembly or combination requirement for two or more components, but only when the component tolerances are equal. In actual practice, some components inherently have smaller natural tolerances than others. However, a preliminarly reading from *Fig.* 44 will give the order of magnitude of the component tolerances as a first approximation. Certain of these can then be increased or decreased to correspond to natural tolerances. A final check on the nomogram will show if assembly requirements can still be met.

By-Product Advantages: Shops in virtually every industry are finding great benefits in process control by use of variables charts. Reduction in scrap and rework and the usual increase in production rates interest them because of the direct financial saving. The statistical addition of tolerances described *cannot* be counted upon to work unless the process is kept in control about the nominal average. Since a control chart does just that, the favorable tolerance accumulations and related gains can be called "by-product" benefits.

About a 40 per cent or greater increase in component tolerances can be allowed without increasing the practical assembly tolerance, when processes are kept in control. This often means less expensive equipment, heavier cuts, or perhaps longer tool life or the like.

As the number of parts that are assembled increases, the relative gain goes up. Moreover, when several parts go together, those with rather small tolerances contribute practically no addition to the net statistical assembly tolerance. Only the parts with the larger tolerances have a significant weight.

A distinct profit will be found in practically every case where selective assembly seems necessary, or where shims or spacers have to be used. These virtually can be eliminated through the use of control charts and the statistical addition.

Most shops know their designers have added tolerance extremes in checking their drawings to see that assembly fits will be satisfactory. Some have taken further advantage of the fact that components in control add more economically and accept certain material beyond specification with only a small chance that the assemblies might be beyond the algebraic fit limits found from adding permissible extremes. A nomogram in the material review section of the shop, where such special deviations are made, along with Lot Plots of the distributions of the component parts make it possible to predict the distribution of the chance assemblies. The distance between lot limits on each Lot Plot serves as the component tolerance on the nomogram.

A New Drawing Notation: For two important reasons each tolerance determined by the recommended statistical method should be indicated by a special marking on the drawing. First, the method always succeeds only when a control chart keeps the distribution free from other than just momentary assignable causes of variation. In any shop, during the period of conversion to this method, the special drawing indication would be a constant reminder to every-

one that the job must not run without a control chart by variables. Then, another logical warning applies to plants that add tolerances statistically on a material review basis to justify the acceptance of material beyond part drawing limits. The special drawing mark would indicate that all such possible allowance has already been made—one should not deviate further using this statistical method, on the assumption that it was not employed in setting up the tolerances originally.

At the Hamilton Standard Div. of United Aircraft Corp., the use of statistical quality control has made factual data available to permit engineering, production engineering, and production to start to work as a unit on these matters of tolerance assignments. Tables compiled from control charts list the natural tolerances against the part number, along with the dimension being made and the equipment-identification tag number. Designers use the table along with the nomogram to compute the most favorable assembly conditions that can be achieved practically.

Whenever this technique is used on a series of accumulating tolerances, each such tolerance is shown on the drawing by this method. It is enough different from any conventional notation that the person reading the drawing has to look further rather than try to guess at its meaning. The following is an example of the special dimensioning used on drawings:

0.5000 (0.0009) See note number —.

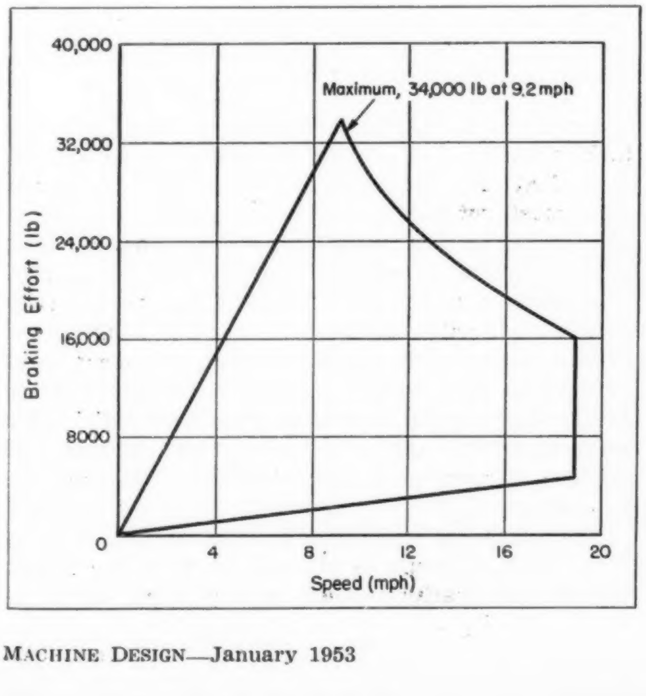
The note number at the bottom of the drawing states, "SQC dimension. Must be run with a control chart." The notation means that this dimension must be held to 0.5000-inch, plus or minus 0.0009-inch maximum variation of individual sizes as controlled by a chart. The first figure, before the parentheses, gives the desired average value. That in the parentheses stands for the maximum allowable three standard-deviation variation from that average. The parentheses, you might say, denote the use of the statistical method.

This method of dimensioning shows that the tolerance has been computed statistically and so is properly economical. The production department should not feel it may need more tolerance. It should be clear that the tolerance actually exceeds the capabilities of the equipment. Production's choice of equipment is guided. Either use the machine that the designer had in mind, a similar one, or one with even a closer natural tolerance (from the control chart records on file).

With this type of control of rough and finished dimensions, forgings and castings often can carry less extra raw material to guarantee proper finishing. But perhaps the most important economy comes from each dimension averaging close to the standard average or nominal, and the parts all coming within specification. The charts keep the operator continually aware from piece to piece of the nature of the variables working, whether they are inherent ones or foreign ones. All these economies and resulting good production practices should assure maintenance of the best possible position in a competitive field.

Locomotive Utilizes Dynamic Braking

BECAUSE it effects a savings in car-wheel and brake-shoe maintenance, dynamic braking has been utilized in a diesel-electric yard switching locomotive built by Baldwin-Lima-Hamilton Corp. for use in lumbering operations. The switcher incorporates a pneumatic speed control which has no transition points—traction motors remain permanently connected in the electrical circuit. Locomotive speed is regulated by traction-motor field shunting and engine governor adjustment through a pneumatic throttle.

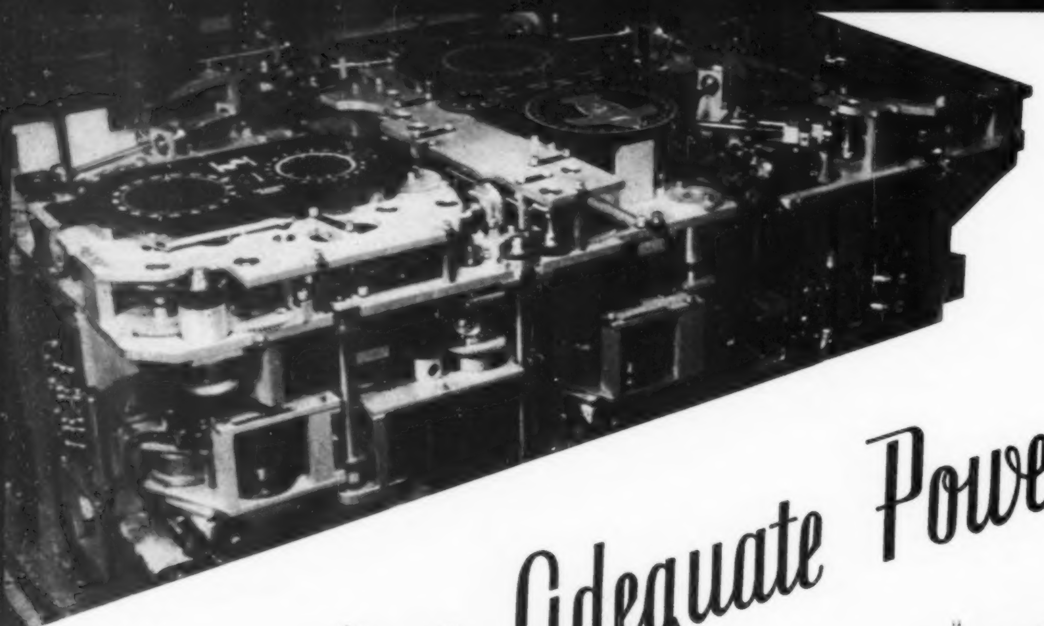


The dynamic braking system consists of a 35-hp motor-driven fan, together with a bank of resistors totaling 1 ohm for each pair of traction motors, plus the necessary electrical control. It is mounted in the engine compartment immediately in front of the cab and electrical control cabinet. Cooling air enters side openings in the hood, passes over the resistors and leaves through the roof. Maximum braking effort of 34,000 lb is attained when the locomotive reaches a speed of 9.2 miles per hour.

The locomotive is a 100-ton, 2-truck, 4-axle unit powered by a 6-cylinder 4-cycle, 800-hp, solid-injection, normally-aspirated diesel engine direct connected to its generator, which feeds the four traction motors.

"The companies most successful in meeting their management manpower requirements will be those which are most successful in getting the best out of the people they already have." LOUIS A. ALLEN, *personnel administration manager, Koppers Co. Inc.*

Providing Adequate Power in Computer



By Milton Felstein
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MANY indicating or computing instruments driven by a motor acting through a gear-train speed reduction represent a measured variable by the angular displacement of a shaft. Such instruments are commonly used where the change in value of the variable with time is dictated by certain dynamic problems or processes which are involved, *Fig.* 1.

An example of such an instrument is a linear potentiometer that is required to represent by its output voltage the distance to an object moving in a known path with known constant speed. The designer of drives for such instruments is faced with the task of selecting a motor and laying out a gear train that will:

a. Be able to produce angular speeds and accelerations of the instrument shaft in excess of those demanded by the change in value of the variable during any problem the instrument is required to handle.

b. Provide a gear-train speed reduction large enough to insure smooth and accurate operation of the instrument.

A simple cut-and-try method for accomplishing this task is presented in this article. Based on a graphical equivalent of requirement *a*, this method, in common with most cut-and-try methods, has the advantage of resulting in straightforward computations in which all the factors affecting the design can be handled. The designer is thus not restricted to such factors as can be handled in direct methods of analysis and he can, in principle, make the study as accurate as he desires. For example, the cut-and-try method can handle easily such factors as gear-mesh efficiencies, speed-dependent and speed-independent friction, and the use of more than one material or thickness for gear wheels.

The graphical technique provides a procedure of "try" that is simple to use and presents the results in an informative visual arrangement. Such an

te Drives

A computer or control mechanism must be capable of keeping up with changes in the variable being followed. To meet this requirement the drive motor must be able to supply sufficient torque to accelerate itself, associated gears, and other moving parts. A straightforward method is presented for determining and comparing velocity-acceleration relationship of the drive and of the function being followed in the problem.

Fig. 1—Gunfire control center on an aircraft carrier. Will the computing mechanism be fast enough to track all possible targets?

Official U. S. Navy photos

arrangement makes it convenient to compare the kinematic capabilities of different drives, as well as to weigh kinematic considerations with mechanical, electrical, servomechanical, factor-of-safety, cost, availability, space, and like considerations involved in the overall design picture.

Even if the designer has his own method of designing motor-and-gear-train combinations to meet the kinematic requisites given above, procedures described here enable him to check the design in a complete study.

Principle of the Method: The curve obtained by plotting instrument-shaft velocities and accelerations producible by the drive will be referred to as the *phase-plane curve of the instrument and drive.* A similar curve for values of velocity and acceleration of the instrument shaft demanded by the change in value of the variable during a given dynamic problem will be called the *phase-plane curve of the problem.* Then, an equivalent statement of requirement *a* is that the phase-plane curve of the instrument and drive must contain (that is, enclose, or lie outside of) the phase-plane curves of all the problems the instrument is required to handle, *Fig.* 2. For the purpose of this technique, one curve contains a second if it lies above and to the right in the first quadrant, above and to the left in the second, below and to the left in the third, or below and right in the fourth.

Only the most demanding problems need actually be plotted, that is, those whose curves lie farthest from the origin. In selecting the most demanding problems, the designer must not neglect to consider demands on the drive such as time allowed for the system to come up to speed or to return to starting position. These factors may prove to be the definitive demands on the drive.

The method discussed here uses phase-plane curves

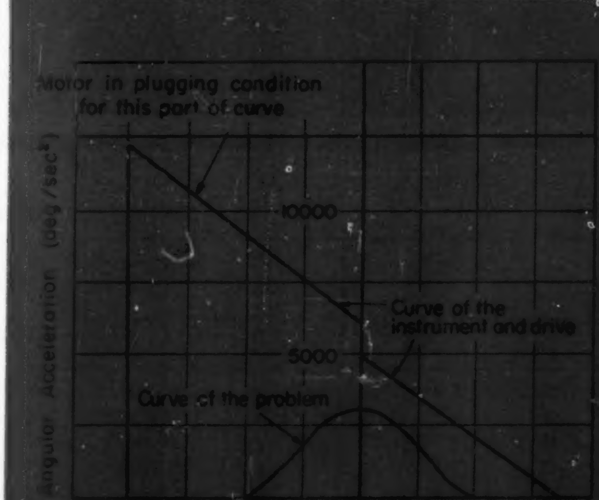

Fig. 2—Phase-plane curves of problem and of instrument and drive. Because the latter curve "contains" the former, it is evident that the drive has adequate capacity to meet the problem conditions

Drive Characteristics

Motor: $J_m = 4.60 \times 10^{-5}$ oz-in.-sec²
Stall torque = 1.3 oz-in.
No-load speed = 5400 rpm
Torque-speed curve: Straight line continuing without a break through range of plugging speeds.

Gears: Material, steel; thickness, 0.13-in.; ball-bearing mounted; coefficient of friction 0.002, efficiency of each mesh, 0.95, static and dynamic. Diameters and moments of inertia are listed in TABLE 1.

Part I: Total effect, speed-independent friction torque, 0.5-oz-in.

Part E: Total effect, speed-dependent friction torque, zero at zero speed, 0.30-oz-in. at 2700 rpm, straight line variation in between.

Instrument: Inertia $J = 15{,}000 \times 10^{-5}$ oz-in.-sec²

Physical characteristics of the system are the same for both directions of turn.

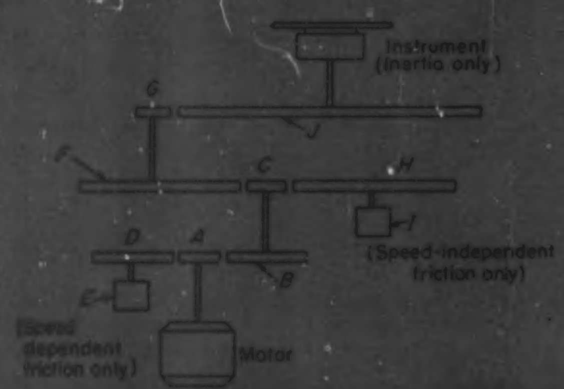

Fig. 3—Schematic arrangement and characteristics of instrument and drive whose performance is indicated in *Fig. 2*

plotted in terms of the angular velocity and acceleration of the instrument shaft. The curves could equally well be plotted in terms of the first and second time derivatives of the variable represented by the displacement of the shaft. This latter scheme is frequently handier, particularly if the phase-plane curves of the problems have been plotted in these coordinates.

Principal steps, which the designer will carry out more or less simultaneously and in rough when starting, are these:

1. Select a motor.
2. Determine the "bracketing values" of the gear-train speed reduction.
3. Select a gear-train speed reduction between these bracketing values.
4. Lay out a gear train to suit this reduction.
5. Compute the phase-plane curve of the instrument and drive using this motor and gear train, and see if this curve contains the phase-plane curves of the most demanding problems.
6. Repeat steps 3, 4, and 5, or steps 1 through 5, as necessary or desired.

Each of the foregoing steps is explained in detail in the following paragraphs, which are correspondingly numbered.

1. SELECTION OF MOTOR. Catalogs of motors commonly give the no-load speed, the stall torque and the moment of inertia of the rotor. Also needed for this study are the torque-speed curve for the condition of maximum control voltage to be used. The usual single-quadrant torque-speed curve covers only the condition in which the motor is turning in the direction of the internal torque of the motor. This curve will be referred to as the "forward-speed" part of the torque-speed curve. If deceleration is to be studied, the torque-speed curve should also cover the condition in which the motor is turning in the

Table 1—Results Independent of Speed*

Part	N	M	Dia.	J	Effective Inertia: Accel. $\frac{J}{N^2e^M}$	Effective Inertia: Decel. $\frac{Je^M}{N^2}$	Speed-independent Friction Torque: Weight	T_f	Accel. $\frac{T_f}{Ne^M}$	Decel. $\frac{T_fe^M}{N}$
Motor	1	0	...	4.60	4.60	4.60	....	....	...	
A	1	0	0.5	0.937	0.937	0.937	0.116	.0002	.0002	.0002
B	2	1	1.0	15.00	3.948	3.563	0.463	.0009	.0005	.0004
C	2	1	0.5	0.937	0.247	0.223	0.116	.0002	.0001	.0001
D	2	1	1.0	15.00	3.948	3.563	0.463	.0009	.0005	.0004
E	2	1	...	....	....	....	....	....	....	....
F	8	2	2.0	240.0	4.155	3.384	1.854	.0037	.0005	.0004
G	8	2	0.375	0.297	0.005	0.004	0.065	.0001	.0000	.0000
H	8	2	2.0	240.0	4.155	3.384	1.854	.0037	.0005	.0004
I	8	2	...	....	....	....	...	.5000	.0693	.0564
J	80	3	3.75	2967	0.541	0.397	6.515	.0130	.0002	.0001
Instr.	80	3	...	15000	2.734	2.009	....	....	....	...
Total	...	...	...	....	25.270	22.064	....	....	.0718	.0584

*Weights and inertias of shafts and bearings neglected. Weights are in ounces, torques in ounce-inches, inertias in 10^{-5} ounce-inch-second². Efficiency (e) is assumed to be 0.95.

opposite direction to the torque—the so-called "plugging" condition. The forward-speed part of the torque-speed curve applies to acceleration of the motor, that is, speeding-up in the same direction the motor is turning, and the plugging-speed part applies to deceleration. The plugging-speed part of the torque-speed curve represents the condition of the motor turning in one direction while full control voltage is being applied to turn it in the opposite direction, which is precisely what occurs during deceleration under full control voltage. Generally speaking, the system will be able to decelerate better than it can accelerate. This is because practically all motors used in servomechanism work have greater torque when opposing the turning than when conforming to it (because of back emf), and also because friction and inefficiency in the system aid deceleration whereas they hinder acceleration. For most motors used in servomechanism work, the forward-speed part of the torque-speed curve can be approximated as a straight line.

This linear approximation is said to be reasonably accurate for induction motors that have high-resistance rotors and are used only in the range up to 55 per cent of synchronous speed. Complete or exact curves should be obtained from the motor manufacturer, measured, or computed, as needed.[1,2,3] Motor "jitter" should also be obtained from the motor manufacturer or measured, if needed.

From the standpoint of acceleration and neglecting friction, when the inertia of the total load is small compared with the inertia of the motor rotor, the ratio of the stall torque of the motor to the inertia of the rotor should be the prime guide in selecting the motor, and the stall torque itself should be the secondary guide. When the inertia of the total load is large compared with the inertia of the rotor, the value of the stall torque should be the prime guide and the torque-to-inertia ratio the secondary guide.[4] The presence of friction will increase the relative importance of the value of the stall torque of the motor.

2. Determination of Bracketing Values of Gear Reduction. Lower bracketing-value of the gear-train speed reduction, N_O, is the lowest N_O that will give smooth and accurate operation of the instrument despite sticking, motor jitter, and other sources of roughness in the drive. The effect of roughness in the drive on the operation of the instrument is reduced as N_O is increased. The lower bracketing-value will also depend on the accuracy required in the instrument setting. Choice of the lower bracketing-value is largely a matter of the designer's judgment, and must take into account the quality of the mechanical workmanship he can expect.

Upper bracketing value of N_O is the largest N_O that will give the maximum speed of the instrument shaft called for by the problems the instrument is required to handle, when the motor has the maximum speed of which it is capable in the presence of expected friction. This maximum speed of the motor is the value on the torque-speed curve that corresponds to the sum of the equivalent friction torques referred to the motor. These friction torques must, of course, be estimated at this stage of the procedure.

3. Selection of Trial Ratio. The first trial N_O is often selected at some preferred value convenient for mechanical design. It may be desired, on the other hand, to make use of some of the more theoretical techniques that have been described for selecting gear-train speed reductions for instrument drives.[4,5,6] Once the designer has selected the first trial N_O, laid

[1] References are tabulated at end of article.

Table 2—Results Dependent on Speed

Applicable data. Speed-dependent friction source, part E: $N = 2$, $M = 1$.
Effective inertia at motor: 25.27 accel., 22.06 decel. (Table 1)
Speed-independent friction torque: 0.072 accel., 0.058 decel. (Table 1)

Motor Characteristics		Speed-dependent Friction Torque at Motor				Kinematics of Instrument			
Speed (rpm)	Torque	Speed of part (rpm)	T_{fs}	$\frac{T_{fs}}{Ne^M}$	$\frac{T_{fs}e^M}{N}$	Effective torque at motor	Accel. or Decel. Motor (rad/sec²)	Accel. or Decel. Inst. (deg/sec²)	Speed of Inst. (deg/sec)
Acceleration									
5400	0.000	2700	0.300	0.158	....	....	....	....	405
3600	0.600	1800	0.200	0.105	....	0.423	1670	1200	270
1800	1.200	900	0.100	0.053	....	1.075	4250	3040	135
0	1.800	0	0.000	0.000	....	1.728	6840	4900	0
Deceleration*									
0	1.800	0	0.000	....	0.000	1.858	8420	6030	0
1800	2.400	900	0.100	....	0.048	2.506	11360	8140	135
3600	3.000	1800	0.200	....	0.095	3.153	14290	10240	270
5400	3.600	2700	0.300	....	0.143	3.801	17230	12340	405

*Motor in plugging condition for lower (deceleration) set of values.

Nomenclature

N_O = Overall gear-train speed-reduction ratio
N = Gear-train speed reduction to a particular component
n = Speed of motor shaft, revolutions per minute
ω = Speed of instrument shaft, degrees per second
α = Angular acceleration (or deceleration) of instrument shaft, degrees per second per second
J = Moment of inertia of a particular component, ounce-inch-second^2
J_m = Moment of inertia of motor rotor, ounce-inch-second^2
J_E = Effective moment of inertia of all components, referred to the motor, ounce-inch-second^2
M = Number of gear meshes between a particular component and the motor
T_f = Friction torque independent of speed, at a particular component, ounce-inches
T_{fs} = Friction torque dependent on speed, at a particular component, ounce-inches
T_m = Torque capacity of motor at a particular speed (from torque-speed curve), ounce-inches
T_E = Effective torque at the motor, ounce-inches
e = Mechanical efficiency at each gear-mesh unit

out the gear train, computed the phase-plane curve of the instrument and drive, and plotted this curve in relation to the phase-plane curves of the problems, it is often possible to proceed rapidly to satisfactory solutions. The first reason for this is that the existing plot will indicate roughly how the new N_O should be selected. The second reason is that, as a rule, the same value for the inertia of the gear train referred to the motor, used in the first try, can be used in the new tries, since this quantity is not sensitive to small changes in N_O. For large changes in the trial N_O as well as for check of the final N_O, a new gear train should be laid out and the computations carried out using a more accurate value of the referred inertia.

4. Layout of Gear Train. A gear train should now be laid out to suit the trial N_O selected. The acceleration of the instrument shaft will be improved if the gear train is designed so as to reduce the inertia referred to the motor.

This is accomplished by properly selecting the number of gear meshes and distributing the overall reduction, N_O, among the meshes to give minimum referred inertia.[6, 7, 8] Reduction of inertia in these ways is possible because, for constant-thickness gear wheels, the referred inertia of a gear wheel varies directly as the fourth power of the diameter and inversely as the square of the speed-reduction ratio to the gear wheel.[9]

5. Computation of Phase-Plane Curve. To compute the phase-plane curve of the instrument and drive for a given direction of turn of the instrument, motor speed is used as independent parameter, and for each value the corresponding values of speed and acceleration or deceleration of the instrument shaft are calculated. In this discussion *acceleration* (speeding-up in the same direction the instrument is turning) is distinguished from *deceleration* (slowing-down in the direction the instrument is turning).

Speed. The speed is obtained from the equation

$$\omega = n\,\frac{360}{60\,N_O} = \frac{6\,n}{N_O} \qquad (1)$$

where terms are as defined in the Nomenclature.

Acceleration and Deceleration. Acceleration and deceleration are obtained from the expression

$$\alpha = \frac{T_E}{J_E}\,\frac{360}{2\,\pi\,N_O} = \frac{57.3\,T_E}{J_E\,N_O} \qquad (2)$$

Effective torque and inertia at the motor, T_E and J_E, must be evaluated. The basic procedure given will apply to acceleration, and changes from this procedure for the case of deceleration will be noted later. The particular mathematical expressions given will apply when the motor torque dominates in accelerating or decelerating the system. Different expressions would apply when this is not the case. Dynamic values of friction torque T_f and efficiency should be used except at speeds near zero.

Torque. Effective torque at the motor, T_E is given by the general expression, in which the signs apply to acceleration:

$$T_E = T_m - \sum \frac{T_f}{N\,e^M} - \sum \frac{T_{fs}}{N\,e^M} \qquad (3)$$

Second term on the right-hand side of the expression is the sum of the respective torques required at the motor to overcome each source of speed-independent friction, taking due account of the number and efficiency of the gear meshes between the motor and the source. Last term in the equation represents the sum of the respective torques required at the motor to overcome each source of speed-dependent friction, taking due account of the number and efficiency of the gear meshes between the motor and the source. The value of T_{fs} will be that corresponding to the speed of the particular source when the motor speed has the value assumed for it as an independent parameter.

For accurate computations the designer may want to include among the speed-independent friction torques those that exist in the bearings due to the weight of the gear wheels and other parts. Such values can be obtained by assuming a suitable coefficient of friction. The labor involved can be reduced by the use of tables or nomograms covering, for instance, the weights of one-inch-long cylinders of various diameters for a particular material. The weight of the motor rotor should not be included if the torque-speed curve has been obtained by measurement. In bearings there also exist friction torques due to the forces being transmitted through the gears. Such torques are proportional to the forces being transmitted and hence are taken account of in this method by the efficiency considerations. Consistent with this, the efficiency values used in the expressions given previously should be those applying to the gear-mesh unit as a whole, not the gear teeth alone.

Inertia. Effective inertia at the motor, J_E, is given by the expression

$$J_E = J_m + \sum \frac{J}{N^2 e^M} \quad \text{(4)}$$

where the second term on the right-hand side represents the sum of the inertias of the gear-train parts, instruments, and other sources of inertia, referred to the motor, taking due account of the number and efficiency of the gear meshes between the source and the motor. This step will involve the computation of the inertias of the gear-train parts. The labor involved can again be reduced by the use of tables or nomograms of inertias of one-inch-long cylinders of various diameters for a particular material.

For deceleration, several changes from the foregoing procedure are involved. Friction and inefficiency aid, rather than hinder the motor, and Equation 3 therefore becomes, for deceleration,

$$T_E = T_m + \sum \frac{T_f e^M}{N} + \sum \frac{T_{fs} e^M}{N} \quad \text{(5)}$$

It should be remembered that for deceleration, the plugging-speed part of the torque-speed curve should be used. For deceleration, further, the value of each inertia source referred to the motor is affected, and Equation 4 becomes:

$$J_E = J_m + \sum \frac{J e^M}{N^2} \quad \text{(6)}$$

Values obtained from the foregoing calculations apply particularly to a given direction of turn of the instrument. The same values will, however, also apply to the reversed direction unless the physical characteristics of the system are different for this direction.

Requirements of the problems to be followed will dictate the particular directions of turn and acceleration or deceleration that will be needed in the study. As described in the next paragraph, each direction of turn and acceleration or deceleration is associated with a particular quadrant of the phase plane.

Plotting the phase-plane curve. Final step is to plot against each other the calculated coexisting values of speed and acceleration or deceleration. A four-quadrant diagram is necessary in order to accommodate all possible conditions of operation. The co-ordinates used will be angular velocity, plotted horizontally, and angular acceleration, plotted vertically. The mechanical quantities speed, ω, and acceleration (speeding-up) or deceleration (slowing-down), α, that are used in the foregoing computation are related to these mathematical co-ordinates as follows: Having established the positive direction of turn of the instrument, when the instrument has this direction of turn, plot acceleration (speeding-up) in the first quadrant and deceleration (slowing-down) in the fourth. When the instrument has the reversed direction of turn, plot acceleration (speeding-up) in the third quadrant and deceleration (slowing-down) in the second.

After the curve has been drawn through the plotted points it should be completed by dropping perpendiculars from the points corresponding to maximum plugging speeds to the axis of velocity. These additional segments are needed to delineate completely the capabilities of the drive. The phase-plane curve of an instrument and drive, for a case requiring the curve in only the first and second quadrants, is illustrated in *Fig. 2*.

Example: An example will illustrate the computation of phase-plane curves and also other procedures described in the foregoing. The example is intended to show method only and values used should not be regarded as a standard of design.

An object can move past an observer in straight-line paths at constant speeds up to 200 feet per second. The minimum cross-over distance is 13 feet. The distance to the object is to be represented by the angular displacement of an instrument shaft, where 1-degree turn of the shaft represents 1-foot distance. It is required to show that the drive described in *Fig.* 3 is kinematically adequate for driving the instrument during any complete trajectory of the object. The instrument might in this case be a helically-wound linear potentiometer in an analog computer where the output voltage of the potentiometer represents the distance.

PHASE-PLANE CURVES OF THE PROBLEMS. The most demanding problem the instrument is required to handle is where the object has maximum speed and minimum cross-over distance. First step is to obtain coexisting values of dD/dt and d^2D/dt^2 that will exist during this problem. From *Fig.* 4,

$$D = \frac{13}{\cos\theta} \quad \text{(7)}$$

$$\frac{dD}{dt} = 200 \sin\theta \quad \text{(8)}$$

$$\frac{d\theta}{dt} = \frac{200\cos\theta}{D} \quad \text{(9)}$$

Differentiating Equation 8 and combining with

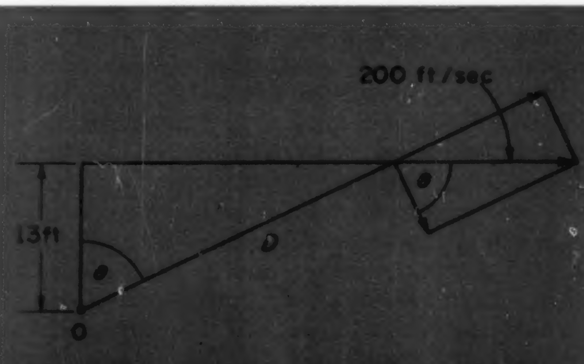

Fig. 4—Diagram illustrating typical problem to be handled by computer drive. Vector represents movement of an object past an observer located at point 0

Equations 7 and 9 gives:

$$\frac{d^2D}{dt^2} = \frac{200^2}{13} \cos^3 \theta = 3077 \cos^3 \theta \quad \text{..........} \quad (10)$$

Use of Equations 8 and 10 yields the following set of values:

θ (deg)	dD/dt (ft/sec)	d^2D/dt^2 (ft/sec²)
90	200	0
80	197	16
60	173	385
40	129	1383
20	68	2553
0	0	3077
−20	−68	2553
−40	−129	1383
−60	−173	385
−80	−197	16
−90	−200	0

Since one degree turn of the instrument shaft represents one foot distance, the second and third columns also give, respectively, in phase-plane co-ordinates, the coexisting values of velocity and acceleration of the instrument shaft in degrees per second and degrees per second per second called for by this problem.

These values when plotted against each other give the phase-plane curve of the problem, *Fig.* 2. It will be noticed that in this case the phase-plane curve of the problem appears only in the first and second quadrants. From the previous explanation of the significance of the various quadrants it is evident that the instrument is required only to speed up when turning in the positive direction (direction of increasing *D* values) and slow down when turning in the negative direction.

PHASE-PLANE CURVE OF INSTRUMENT AND DRIVE. Next step is to compute the phase-plane curve of the instrument and drive in order to determine the ability of the drive to speed up the instrument when turning in the positive direction and slow it down when turning in the negative direction. In this particular example, the physical characteristics of the system are the same for both directions of turn, hence no distinction need be made between the directions of turn in the computation. The computation is shown in TABLES 1 and 2. The calculated values are plotted in the first and second quadrants of *Fig.* 2. The kinematic adequacy of the drive is indicated by the fact that the curve of the instrument and drive "contains" the curve of the problem. The calculated values also represent the ability of the drive to speed up the instrument when turning in the negative direction and slow it down when turning in the positive direction. The applicable plot would appear in the third and fourth quadrants and would look like the present curve of the instrument and drive rotated 180 degrees about the origin as center.

Acknowledgement is due Mr. Macon Fry, a member of the staff of the Operations Research Office, War Department, for the idea of checking the kinematic adequacy of instrument drives by means of phase-plane curves. Valuable advice in the preparation of the article was obtained from Mr. F. J. Miller and Mr. F. A. Ross. both of the Naval Gun Factory.

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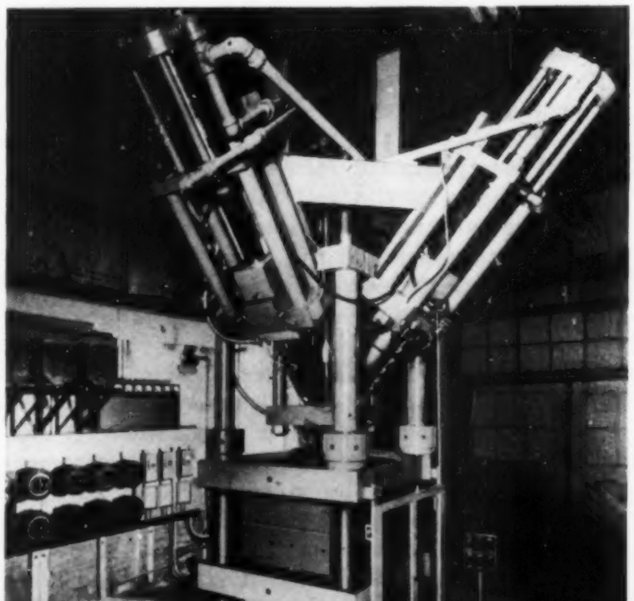
Molder Has Large Capacity

HAVING a capacity of 22 pounds of powder or polystyrene per shot, this vertical molding machine, designed and developed by Worcester Moulded Plastics Co., has a 25 per cent greater molding capacity than similar machines. Four huge preplasticizing units at the top of the machine preheat the



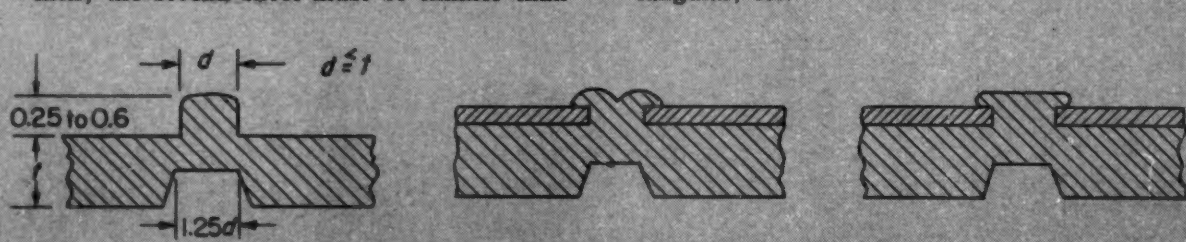
material to give it greater flowing properties which, together with 25 per cent greater locking pressure on the dies, permit the molding of pieces in sizes heretofore impossible. Size of the platen is 42 by 60 inches; the machine has a 40-inch stroke; die space is 40 inches; and average time required to complete a molding cycle is between 1 and 2 minutes.

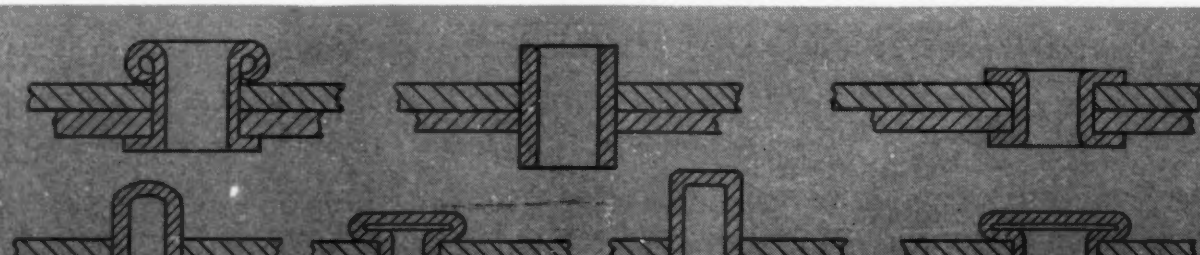
Joining Stampings

Methods of designing economical joints for multiple-piece stamping assemblies

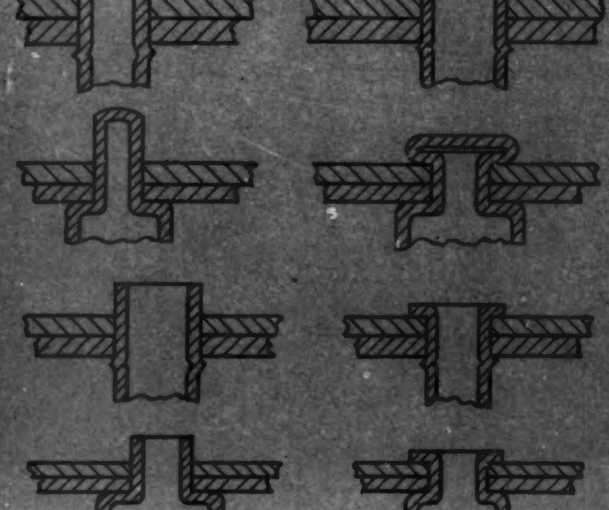
By Federico Strasser
Santiago, Chile

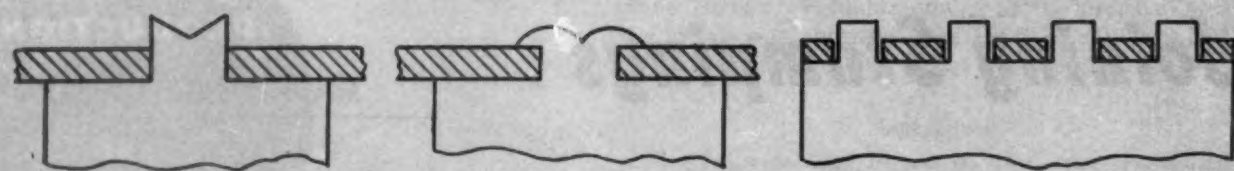
By means of a comparatively simple and inexpensive operation, a small portion of the metal may be extruded and utilized as a rivet. This operation is done by incomplete punching, and to facilitate the operation, the punch depression is made somewhat larger in diameter than the extruded boss. Limitations are that it can be done only with comparatively thick stock, not less than 0.1-inch; the second sheet must be thinner than the embossed part, by at least 50 per cent, because the length of the boss is generally limited to a fraction of the stock thickness; and because of the shortness of the boss it is not possible to make a regular rivet-head, only a little spreading or flattening can be done. The cross-section of the extruded boss is generally round, however for assembly purposes it can be made oval, square, rectangular, etc.



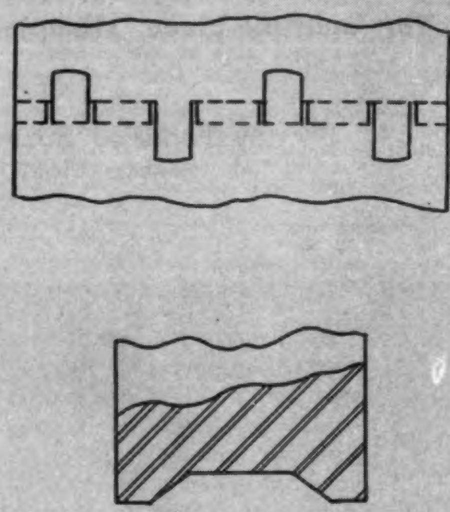


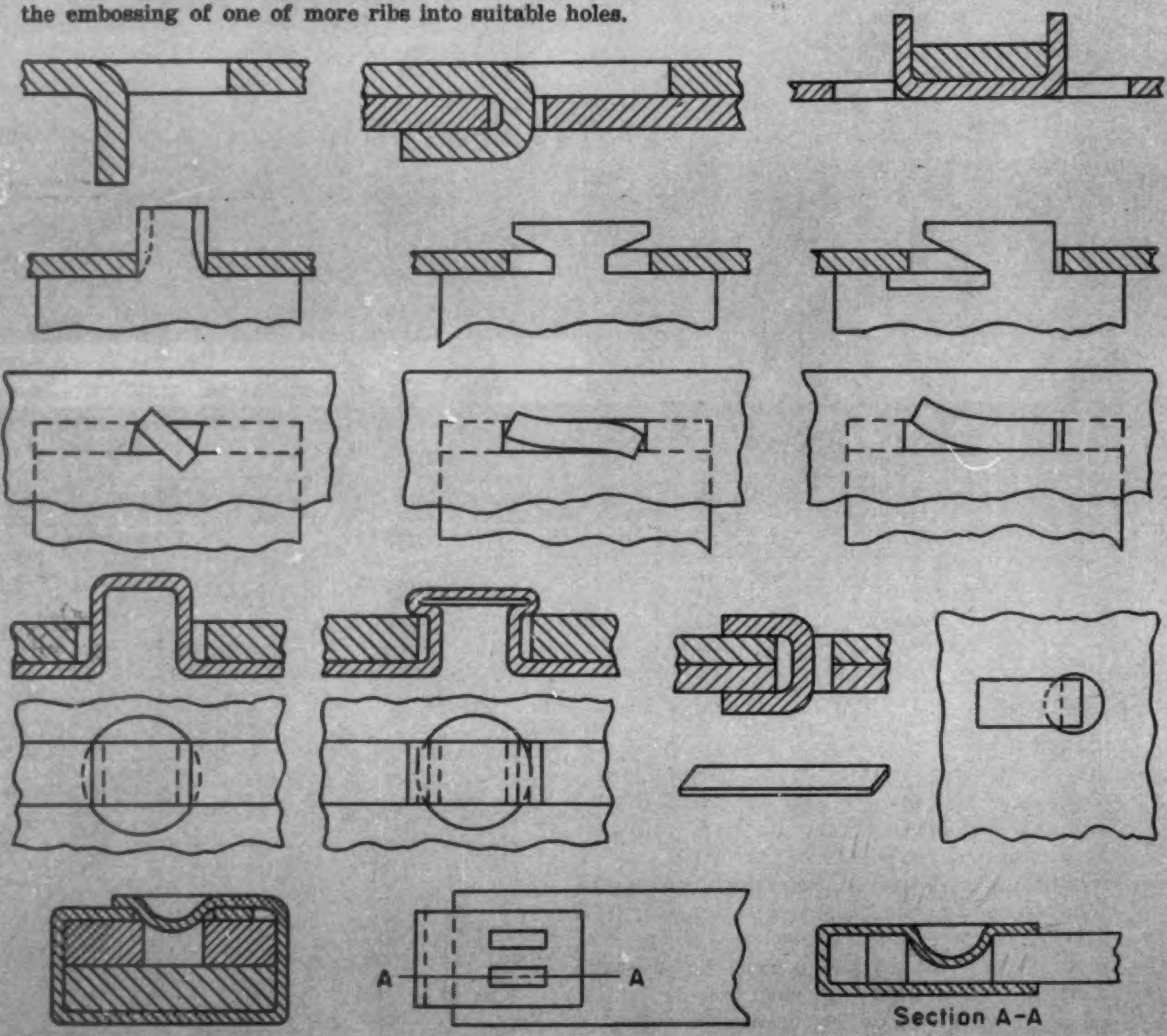
Hollow rivets in different forms may be employed. A short piece of tubing may be flared and spread flat on both ends, may be flared at one end and curled at the other, or curled at both ends. In lieu of tubing, the rivet may be constituted by a flanged shell which is flattened for fastenening. Shells may be fastened against stampings by making a bead or reducing the end of the shell at the proper distance and flattening the free end. A shell may be fastened to a stamping on its open end much the same as tubing.

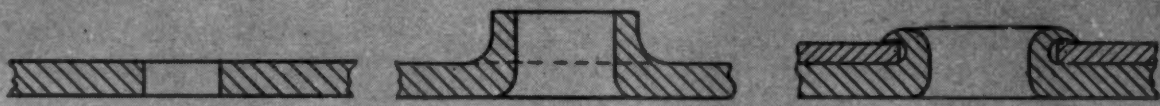




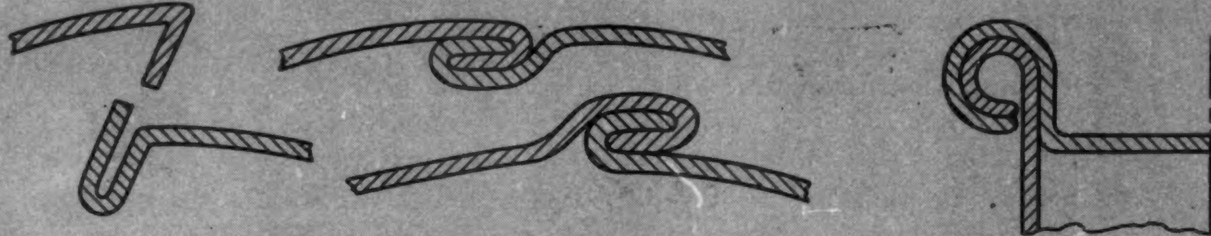
A good press operation for assembling stampings is folding small lugs in the proper places so as to obtain a locking action. The simplest form is a single lug located in a longitudinal hole and opened sidewise. If the stampings are made from thin stock and are large enough, several lugs may be used. For center lugs, forming tools can be used which cut and bend the lugs simultaneously. These are then introduced in the corresponding longitudinal holes of the counterparts and folded flat. If the locking must be tight, the lugs should be twisted. Tapered lugs are somewhat more expensive, but on the other hand, more efficient. Thin and narrow strips are best assembled to stampings or sheets by first forming U-shape sections which can be introduced into the proper holes and flattened. A piece of steel sheet which fits snugly into holes in the stampings can be used for riveting. For light duty, this type of economical rivet gives surprisingly good performance. Another good light duty device is the embossing of one of more ribs into suitable holes.



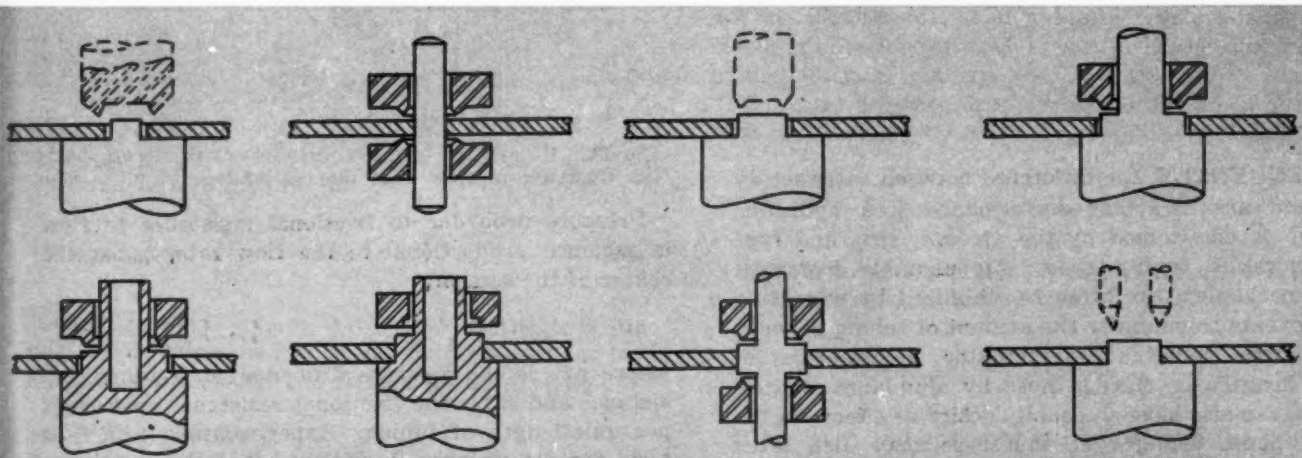




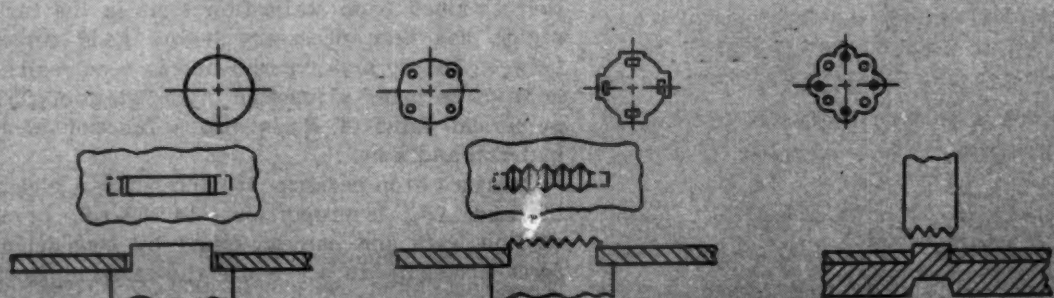
Successful riveting with thin or medium-thick stock can be obtained with tubular extrusions. With this method a punched hole is partly transformed into a drawn boss. After assembly the hollow boss is flared or spread outwards.



If the shape of the workpiece allows it, a seaming operation can be used for joining two straight edges. The seam may be internal or external. Two drawn shells or a shell and a cover may also be joined together tightly by curling.



Locking together of two parts can often be accomplished best by staking—expanding the metal where a shaft or lug fills a hole. This is done by either squeezing in the metal around the hole or spreading out the shaft. Generally the staking is not made all around the hole, but is interrupted; in most cases it is done only at few points on the periphery as shown in the accompanying sketches which represent some typical examples. Staking is not limited only to round machined parts. The shaft portion may have any desired shape and, where practicable, may be simply extruded.



Electric Analogies for Hydraulic Analysis

Part 2—Tubing Characteristics

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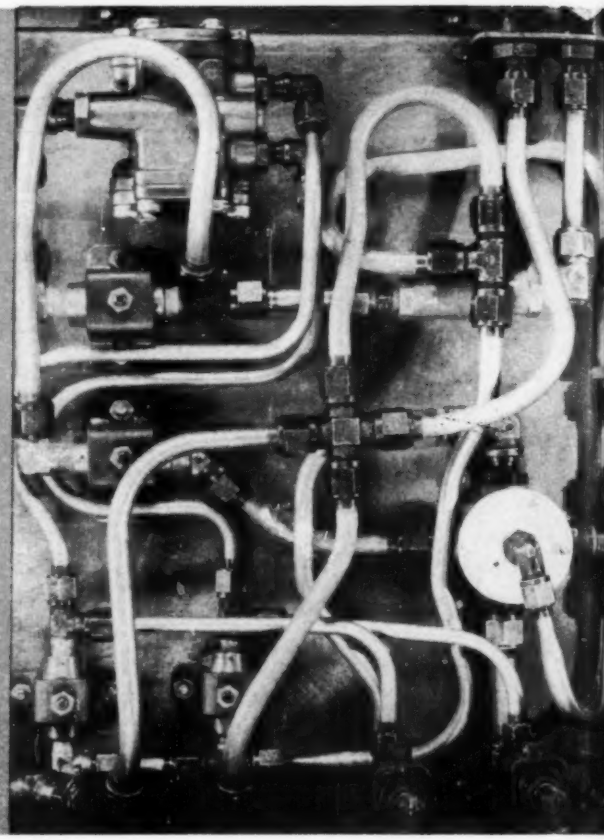

FREQUENTLY, the difference between satisfactory and unsatisfactory performance of a hydraulic system is determined by the amount, size, and type of hydraulic tubing used. An unstable hydraulic servomechanism can often be stabilized by relocating components to minimize the amount of tubing between the control valve and the actuator. Replacement of short lengths of flexible hose by aluminum or steel tubing can also have an equally desirable effect. On the other hand, changes of this type may also often have an undesirable effect. In any case, the dynamic characteristics of hydraulic tubing influence to a large extent the dynamic behavior of the hydraulic system and must be considered in the system design.

Equations of Fluid Transmission: The derivation that follows applies to straight, continuous sections of nonflexible metal hydraulic tubing and is based on the assumption that there is no variation in pressure along the tubing cross section. An incremental section of tubing of length Δx is shown in *Fig.* 7. The pressure P_2 differs from the pressure P_1 because of:

1. Pressure drop due to frictional resistance to flow.
2. Pressure drop necessary for acceleration of the fluid mass.

Similarly, the output flow rate q_2 differs from the input flow rate q_1 because of:

1. Decrease in flow rate due to compressibility of the fluid and elasticity of the tubing.
2. Decrease in flow rate due to leakage.

Pressure drop due to frictional resistance to flow is assumed proportional to the flow rate, q_0, at the center of the section;

$$\Delta P_f = R_x q_0 (\Delta x) \qquad (27)$$

where ΔP_f is the pressure drop due to frictional resistance and R_x is the frictional resistance coefficient per unit length of tubing. Experimental static flow tests for determining R_x indicate that the resistance coefficient is substantially a constant for both the laminar and turbulent regions of flow. However, the turbulent value is considerable higher than the laminar value. In the transition region, R_x is definitely not constant and is difficult to determine. For dynamic studies, the assumption of a constant resistance coefficient has proved quite adequate although adjustment of the numerical value to twice that obtained from static flow tests in the turbulent region has been necessary before good correlation between theory and experiment has been realized. In addition to being a function of the state of flow, the numerical value of R_x is also a function of tubing material and size.[1]

Pressure drop necessary to accelerate a given mass of fluid, ΔP_m, is proportional to the rate of change of flow rate and can be found by considering the force necessary to accelerate the mass of fluid con-

[1] References are tabulated at end of article.

IN BEHAVIOR a hydraulic line is essentially comparable to an electric transmission line. Similarities in fluid and current transmission provide a basis for the derivation of representative two-terminal network electric analogies. In this article, second of a series, the analogous characteristics of hydraulic tubing are discussed, supplementing the analogy techniques presented in Part 1 for basic system components. Application of these techniques to the analysis of a hydraulic servomechanism will be demonstrated in the final article.

tained in the incremental volume of *Fig.* 7. From Newton's Law this force is

$$F = ma = m\frac{dv}{dt} \qquad (28)$$

where F is the force, m is the fluid mass, a is the fluid acceleration, and v is the fluid velocity at the center of the section. The fluid mass is given by

$$m = \rho V = \rho A(\Delta x) \qquad (29)$$

where ρ is the fluid mass density, V is the fluid volume, and A is the tubing cross sectional area. Also,

$$\frac{dv}{dt} = \frac{1}{A}\frac{\partial q_0}{\partial t} \qquad (30)$$

Substitution of Equations 29 and 30 into Equation 28 yields

$$F = \rho(\Delta x)\left(\frac{\partial q_0}{\partial t}\right) \qquad (31)$$

Dividing Equation 31 by A gives the required pressure drop as

$$\Delta P_m = \frac{F}{A} = \frac{\rho(\Delta x)}{A}\left(\frac{\partial q_0}{\partial t}\right) \qquad (32)$$

Decrease in flow rate due to fluid compressibility and tubing elasticity can be derived from the definition of the fluid compressibility constant β, which is the fractional change in volume per unit change in pressure. For tubing, this compressibility constant is modified to include the effects of tubing elasticity. Therefore, if $(\Delta A)(\Delta x)$ is a small element of volume within the tubing

$$\beta = \frac{\frac{\Delta V}{V}}{\Delta P_0} = \frac{\Delta A(\Delta x)}{A(\Delta x)}\left(\frac{1}{\Delta P_0}\right) \qquad (33)$$

where ΔP_0 is the change in pressure at the center of the incremental section. From Equation 33

$$\Delta V = \Delta A(\Delta x) = \beta A(\Delta x)(\Delta P_0) \qquad (34)$$

Differentiation of Equation 34 with respect to time yields

$$\Delta q_c = \frac{d(\Delta V)}{dt} = \beta A(\Delta x)\left(\frac{\partial P_0}{\partial t}\right) \qquad (35)$$

where Δq_c is the flow rate decrease due to effects of fluid compressibility and tubing elasticity.

Decrease in flow rate due to leakage, Δq_G, is assumed proportional to the pressure at the center of the section, P_0, and can be written as

$$\Delta q_G = G_x P_0(\Delta x) \qquad (36)$$

where G_x is the leakage factor. For straight sections of metal hydraulic tubing, G_x is essentially zero and is not usually considered in hydraulic system analyses. However, the concept will be useful in the discussion of the electrical analogy.

The total pressure drop across the incremental section can be found by combining Equations 27 and 32:

$$P_1 - P_2 = R_x q_0(\Delta x) + \frac{\rho}{A}(\Delta x)\left(\frac{\partial q_0}{\partial t}\right) \qquad (37)$$

By combining Equations 35 and 36 the net decrease in flow rate through the section is

$$q_1 - q_2 = G_x P_0(\Delta x) + \beta A(\Delta x)\left(\frac{\partial P_0}{\partial t}\right) \qquad (38)$$

If pressure and flow rate at any point in the tubing

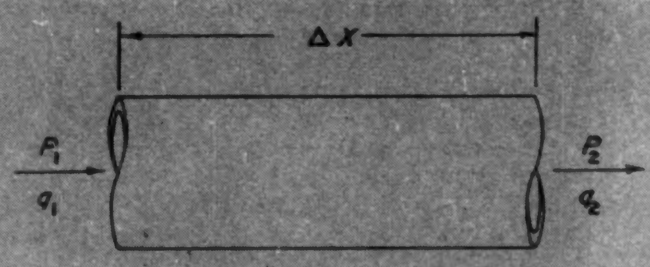

Fig. 7—Analysis of pressure and flow rate differences for incremental section of tubing provides basis for derivation of basic fluid transmission equations

are assumed to be continuous functions of the displacement along the tube at any given time, these quantities can be expressed in expanded form as a set of infinite series. Thus, at any point along the incremental section of *Fig.* 7 the pressure, *P*, is given by

$$P = P_0 + x\left(\frac{\partial P_0}{\partial x}\right) + \frac{x^2}{2!}\left(\frac{\partial^2 P_0}{\partial x^2}\right) + \cdots \quad (39)$$

and the flow rate by

$$q = q_0 + x\left(\frac{\partial q_0}{\partial x}\right) + \frac{x^2}{2!}\left(\frac{\partial^2 q_0}{\partial x^2}\right) + \cdots \quad (40)$$

By substituting $x = \pm \Delta x/2$,

$$P_1 = P_0 - \frac{\Delta x}{2}\left(\frac{\partial P_0}{\partial x}\right) + \frac{1}{2!}\left(\frac{\Delta x^2}{4}\right)\left(\frac{\partial^2 P_0}{\partial x^2}\right) + \cdots \quad (41)$$

$$P_2 = P_0 + \frac{\Delta x}{2}\left(\frac{\partial P_0}{\partial x}\right) + \frac{1}{2!}\left(\frac{\Delta x^2}{4}\right)\left(\frac{\partial^2 P_0}{\partial x^2}\right) + \cdots \quad (42)$$

and

$$q_1 = q_0 - \frac{\Delta x}{2}\left(\frac{\partial q_0}{\partial x}\right) + \frac{1}{2!}\left(\frac{\Delta x^2}{4}\right)\left(\frac{\partial^2 q_0}{\partial x^2}\right) + \cdots \quad (43)$$

$$q_2 = q_0 + \frac{\Delta x}{2}\left(\frac{\partial q_0}{\partial x}\right) + \frac{1}{2!}\left(\frac{\Delta x^2}{4}\right)\left(\frac{\partial^2 q_0}{\partial x^2}\right) + \cdots \quad (44)$$

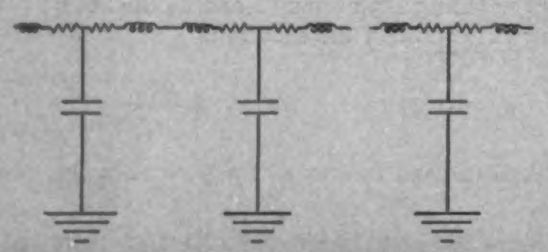

Fig. 8—Cascaded "T" network representation of electric transmission line can be used to approximate behaviour of hydraulic line

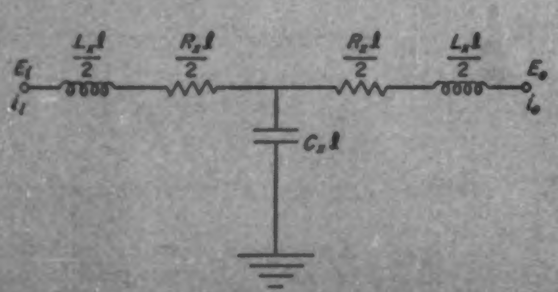

Fig. 9—For tubing lengths of 25 feet or less the single "T" network analogy shown can be used for most hydraulic systems

Table 2—Analogous Electric-Hydraulic Quantities for Hydraulic Tubing

Tubing Parameter	Electric Analogy	Correlation
Pressure, P	Voltage, E	$E = P$
Flow rate, q	Current, i	$i = q$
Fluid compressibility and tubing elasticity per unit length, βA	Capacitance per unit length, C_x	$C_x = \beta A$
Frictional resistance coefficient per unit length, R_x	Resistance per unit length, R_x	$R_x = R_x$
Equivalent fluid inertial constant per unit length, ρ/A	Inductance per unit length, L_x	$L_x = \rho/A$
Tubing leakage factor per unit length, G_x	Conductance per unit length, G_x	$G_x = G_x$
Tubing length, l	Transmission line length, l	$l = l$

Neglecting terms of higher order differentials and subtracting Equation 42 from Equation 41,

$$P_1 - P_2 = -\Delta x\left(\frac{\partial P_0}{\partial x}\right) \quad (45)$$

Also, subtracting Equation 44 from Equation 43,

$$q_1 - q_2 = -\Delta x\left(\frac{\partial q_0}{\partial x}\right) \quad (46)$$

By combining Equations 45 and 46 with Equations 37 and 38 the fluid transmission equations for straight sections of the hydraulic tubing are obtained.

$$\frac{\partial P_0}{\partial x} + R_x q_0 + \frac{\rho}{A}\left(\frac{\partial q_0}{\partial t}\right) = 0 \quad (47)$$

$$\frac{\partial q_0}{\partial x} + G_x P_0 + \beta A\left(\frac{\partial P_0}{\partial t}\right) = 0 \quad (48)$$

Electric Transmission Line Analogy: The procedure used for deriving the equations of fluid transmission parallels almost exactly the procedure used for deriving the equations which describe the behavior of an electric transmission line.[2] These analogous equations are

$$\frac{\partial E}{\partial x} + Ri + L\left(\frac{\partial i}{\partial t}\right) = 0 \quad (49)$$

$$\frac{\partial i}{\partial x} + GE + C\left(\frac{\partial E}{\partial t}\right) = 0 \quad (50)$$

A comparison of Equations 47 and 49 and Equations 48 and 50 reveals that they are identical in form and thus serve as the basis for using the characteristics of the electric transmission line in describing the behavior of hydraulic tubing. Analogous quantities in these equations are tabulated in TABLE 2.

Solution of the transmission line equations for frequency response characteristics yields two parameters which completely determine the behaviour of the transmission line. These parameters are the characteristic impedance, Z_o, and the propogation function αl, which are defined by

$$Z_o = \sqrt{\frac{R_x + j\omega L_x}{j\omega C_x}} \quad (51)$$

$$\alpha l = l\sqrt{(R_x + j\omega L_x)(j\omega C_x)} \quad (52)$$

where l is the length of the transmission line, ω is the frequency of operation in radians per second and j, the operator, is equal to $\sqrt{-1}$. Since the leakage factor for hydraulic tubing is negligible it is not included in these defining equations.

By treating hydraulic tubing as an electric transmission line, a very important simplifying approximation can be obtained. From network theory, a series of cascaded "T" sections, *Fig.* 8, approximate line behavior. The error in the characteristic impedance of the simulated line is given by

$$\delta(Z_o) = 100\left(\frac{1}{8}\right)\left(\frac{\alpha l}{n}\right)^2$$

$$= \frac{100}{8}(R_x + j\omega L_x)(j\omega C_x)\left(\frac{l}{n}\right)^2 \quad (53)$$

Error in the propogation function is a third of the error in Z_o. In Equation 53, n represents the number of "T" sections used for the approximation. Thus the distributed parameters of the hydraulic tubing can be replaced with the lumped parameters of the simulating network.

In most hydraulic system applications, tubing lengths of 25 ft or less can be satisfactorily approximated with the single "T" network shown in *Fig.* 9. For longer lengths of tubing or when the system resonant frequencies become greater than 20 cps two or more sections may be required for satisfactory simulation.

Bends and Fittings: The transmission line analysis given here applies only to straight sections of uniform hydraulic tubing. When bends and fittings occur they must be handled as entirely separate components in the electric analogy. Therefore, a long piece of tubing in which a single fitting occurs must be simulated with two sections of "T" networks separated by the impedance of the fitting. This impedance is generally a resistance. When bends occur in the tubing a similar situation exists and the electric analogy will have the "T" networks separated with impedance representations for the bends. Unless the radius of the bend is very large this impedance can also be considered resistive. Experimental resistance characteristics obtained for various bend angles of a section of aluminum tubing are shown in *Fig.* 10.

Flexible Hose Characteristics: Attempts to treat sections of flexible hose in the same manner as metal tubing have not been successful. Discrepancies which have occurred have recently been traced to the fact that the fluid compressibility and tubing elasticity factor per unit length of hose is not a constant but a very sensitive function of pressure along the hose. The assumption that this factor is an evenly distributed parameter along the tubing is fundamental to the

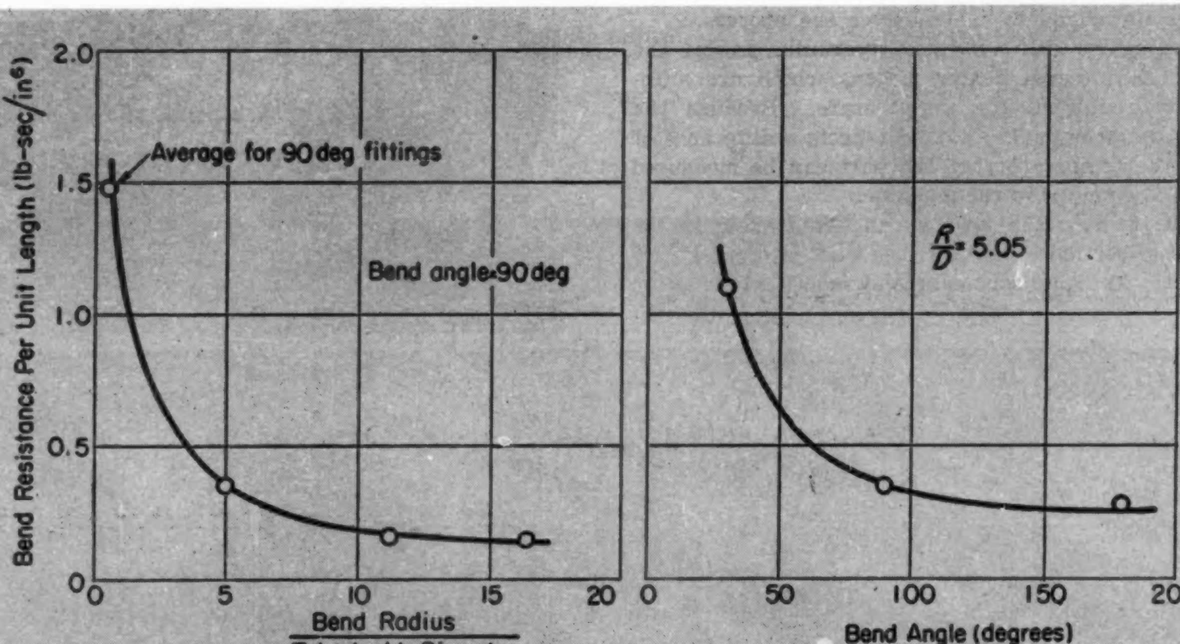

Fig. 10—Resistance characteristics curves show experimental results obtained with a section of aluminum tubing

transmission line analysis and therefore cannot be valid in applications to flexible hose.

Although the transmission line analysis breaks down in systems containing flexible hose the qualitative effect can be stated. In the electrical analogy the same qualitative effect can be obtained by adding increased capacitance to the transmission line. Therefore, if a system is analyzed on the basis that metal tubing is used, the effect of adding increased amounts of capacity to the line will determine whether flexible hose will improve or impair system performance.

Hydraulic-electric analogies developed in this and the preceding part provide a basis for analysis of a complete hydraulic system. In the next and concluding part of this series, application of these analogies to the analysis of a hydraulic servomechanism will be demonstrated.

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1. James E. Campbell—"Investigation of the Fundamental Characteristics of High Performance Hydraulic Systems," *USAF TR 5997-Restricted*, June, 1950.
2. E. A. Guillemin—*Communication Networks*, Vol. II, John Wiley & Sons Inc., New York, 1935.

Bends Aluminum Alloy Wing Beams

USED primarily for bending wing beams but capable of handling other type beams, this new machine can bend a 5 by $3\frac{1}{2}$-inch 75ST aluminum alloy beam as much as 50 degrees. It can handle beams with an overall length of 60 feet or more. Insert assemblies are forced together by two upper cylinders and locked in place by two horizontal pistons. Parts are heated to 325 F at the point of bend. A rotational control can be adjusted so that springback does not warp the bent part out of the proper plane.

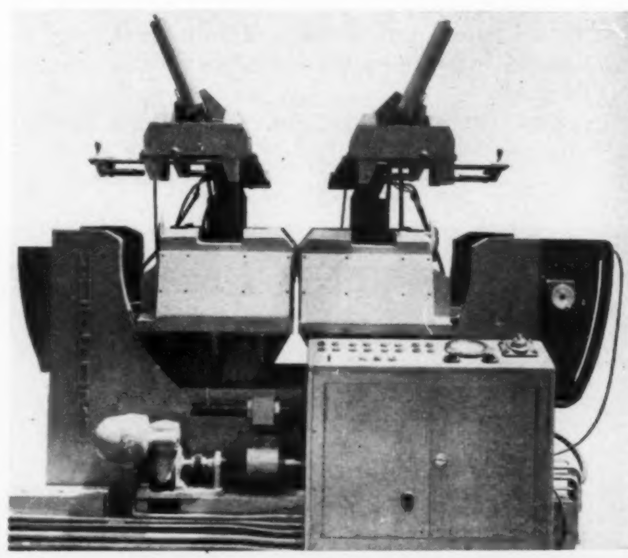
Prior to the development of the new beam bender the bending of beam caps and other extrusions had been accomplished by the use of bend blocks and dies. The 75ST alloy, however, requires hot bending, making previous methods impractical. This new machine accomplishes in 3 minutes jobs which formerly required 30 minutes, and greater accuracy in bending is claimed.

Designed by the manufacturing research branch of Lockheed Aircraft Corp. and built by Hufford Machine Works Inc., the machine incorporates tooling which provides for complete control over the process.

Operating on self-contained hydraulic power, the bender has integral heater platens which are automatically regulated for temperature. Because the amount of springback obtained reflects a difference of ± 2 F, the temperature of the part can be measured while it is clamped in the machine.

Inserts fit into the jaws which adapt them to the part cross-section; clamps close the jaws and grip the part. An equalizer ensures symmetrical elevation of the jaws when bending, and adjustable centers of motion for each jaw, as well as automatic, adjustable stops, control the amount of bend. The jaws rotate around the beam center line to proportion the dihedral and sweepback angles properly and to allow for springback in both planes.

The machine can pull backward to correct a beam which is inadvertently overbent during initial adjustment, and it can also remove twist warpage which sometimes occurs in machined or extruded parts. Speed of bending is adjustable and all functions are controlled by pushbuttons on the control stand.



Improving Fatigue Life

Tests on roller chain link plates show comparative effects of several methods of inducing beneficial residual stresses

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INCREASING severity of service conditions in machinery is prompting constant search for lighter, stronger, and less expensive elements to do a required job. This has given impetus to many research programs aimed at developing parts which are more fatigue-resistant under heavy or abnormal loads.

As an example, roller chain has been the subject of considerable research along these lines. The findings reported in this article, although specifically pointed toward the development of improved chain links, have significance in relation to many other types of machine elements employing similar configurations and processing methods.

In normal use, recommended loads on chain drives are well below the ultimate strength of the chain. However, loads greater than recommended are sometimes demanded, and in these instances fatigue failure of the link plate may occur due to the nature of the loading. The constant cycling of the load from a maximum tensile load to a minimum, or zero tensile

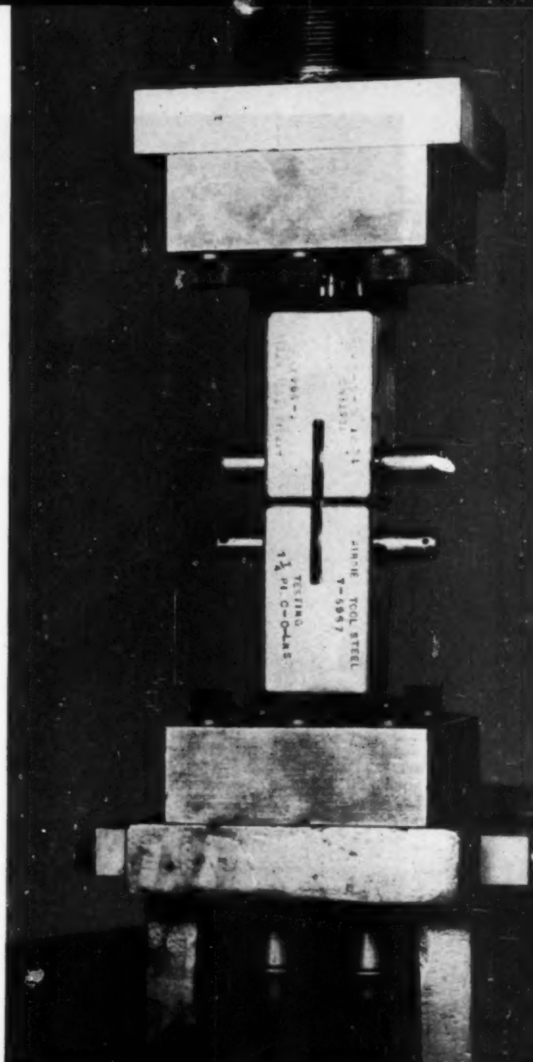
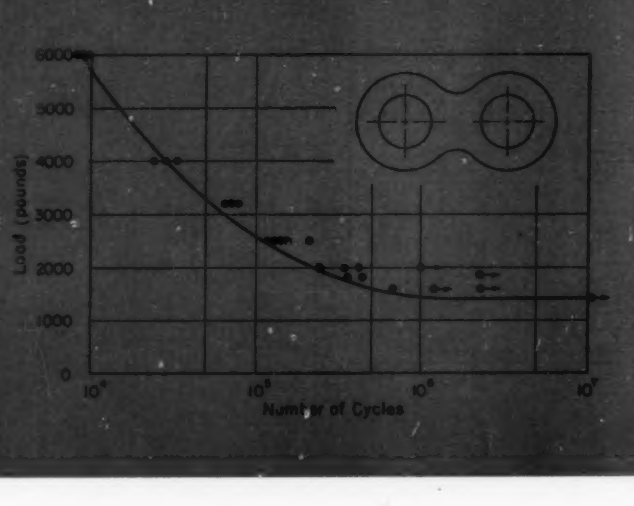
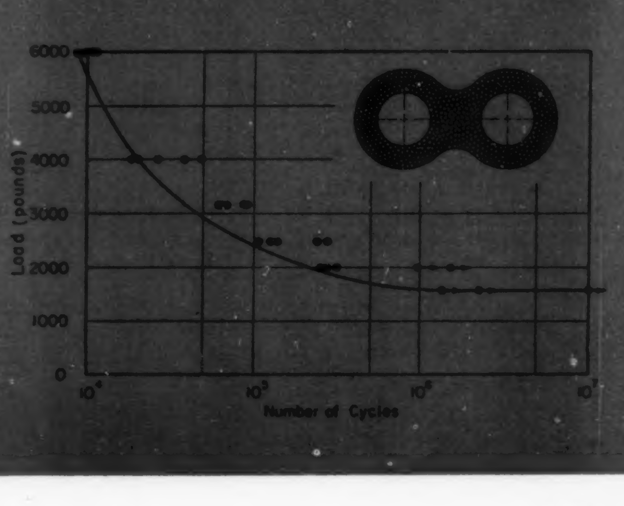


Fig. 1—Above—Link plate mounted in fatigue-testing machine. Effects of several alternative residual-stress treatments on endurance are shown in the succeeding illustrations

Fig. 2—Below, left—L-N curve for standard production link used as control sample

Fig. 3—Below—L-N curve for shot-peened link plate





load, coupled with stress concentrations and impact loads inherent in roller chain design, all contribute to possible fatigue failure at high loads.

Various changes in the construction of roller chain have been tried in order to improve the fatigue characteristics. For example, heavy-series chain with thicker link plates and multiple-strand chain having substantial press fits between pins and center link plates reflect previous effort in this direction.

In investigating other avenues for improvement, research has now been focused on the treatment of the chain link plates by various means. These treatments are applied to otherwise finished parts, the object being to impart beneficial residual stresses. All of these variations of a long-recognized principle have been used in various other mechanical parts subject to cyclic stressing.

Tests were run on individual link plates for No. 100 ASA roller chain rather than on assembled chain. The variations in fatigue life of assembled chain, caused by factors such as bending, eccentric loading, impact, and minor differences in pitch due to manufacturing methods, are eliminated when the part is tested alone. Testing individual link plates permitted judgment of the efficacy of a particular method of treatment with respect to fatigue resistance of the part. However, it was not expected that the improvement in the link plate would necessarily show equal improvement in the chain.

A Sonntag universal fatigue-testing machine was used for the tests, the desired tensile load being applied and released at the rate of 1800 cycles per minute. A small preload was applied to the specimen so that when the main load was released, the sample was under slight tensile load. *Fig.* 1 shows a link plate mounted in the fatigue machine.

All tests reported here were run on the same size pitch link plates since only the effect caused by the variation in treatment was being studied. Most of the plates were tested to failure, but several were removed from the test machine unbroken. The results of the latter tests have been marked with an arrow on the *L-N* curves, *Figs.* 2 through 8.

Seven groups including a control sample were tested. Treatments, indicated in the caption for each L-N

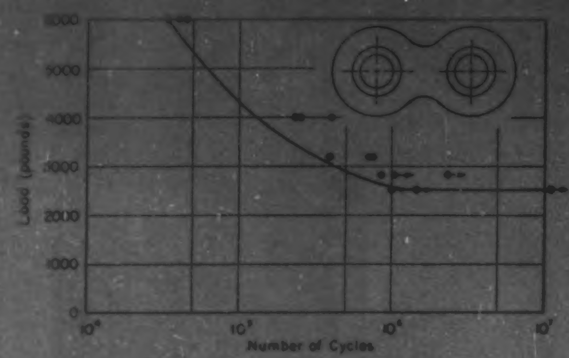


Fig. 4—*L-N* curve for link plate having bushed holes with standard controlled press fit

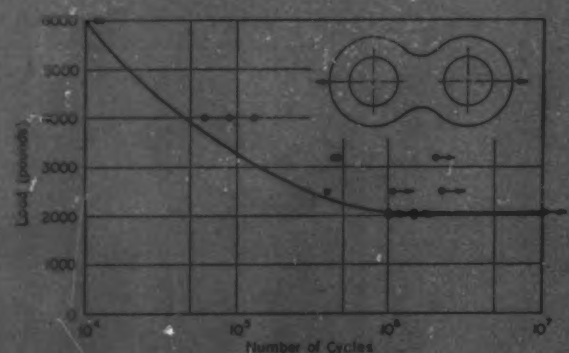


Fig. 5—*L-N* curve for link plate prestressed in tension

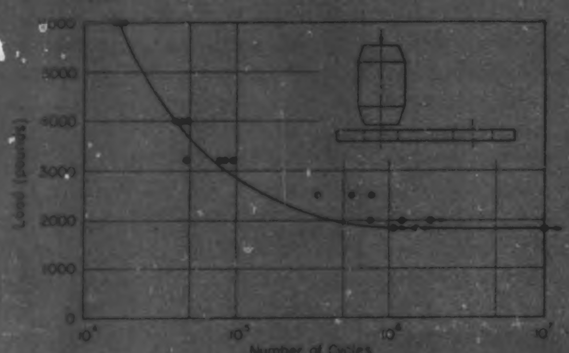


Fig. 6 — Above — *L-N* curve for link plate with holes drifted by an oversize pin

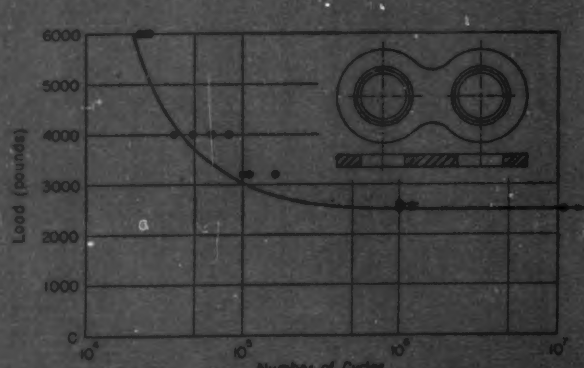


Fig. 7—*L-N* curve for link plate with embossed ring around hole

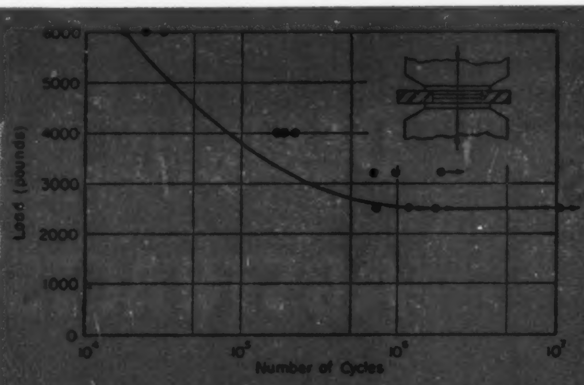


Fig. 8—Above—*L-N* curve for link plate with narrow compressed band adjacent to hole

Fig. 9—Below—Examples of fatigue failure of a roller link plate

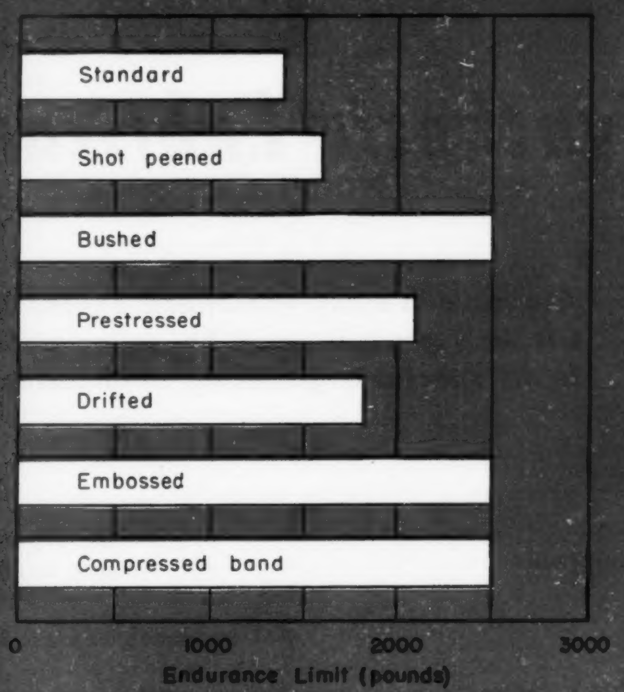
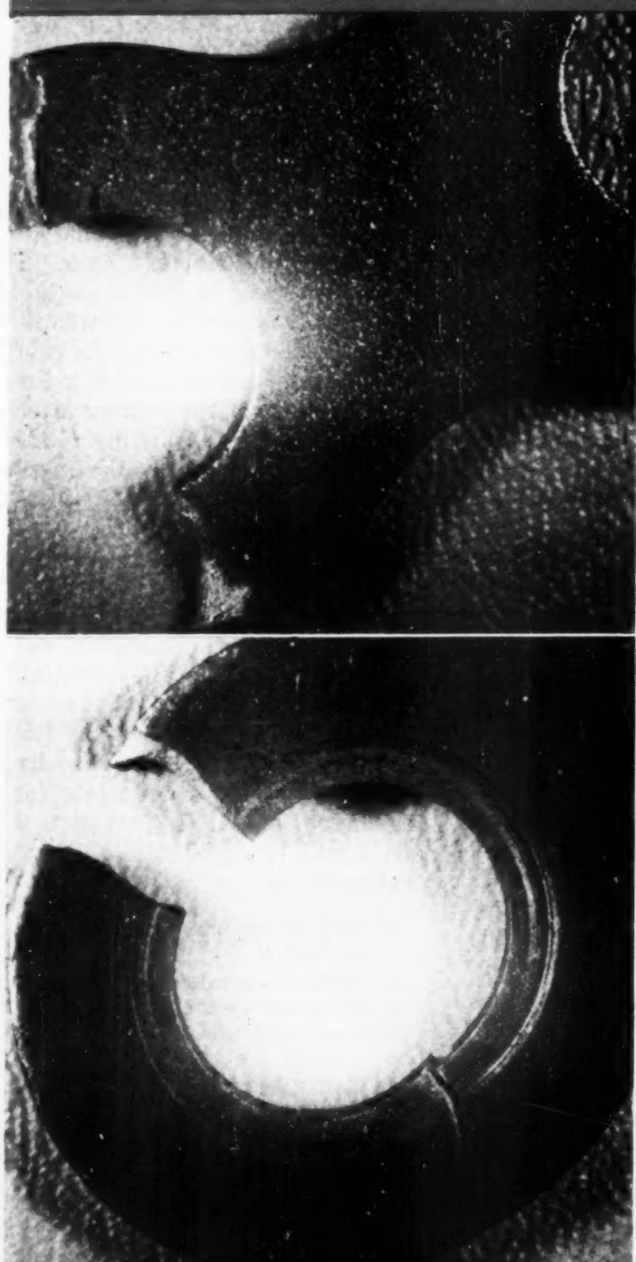


Fig. 10—Comparison of endurance limits of link plates with various methods of treatment



curve chart, were applied to prestress the area of the hole where fatigue failure of roller link plate normally begins. This type of failure is shown in *Fig.* 9.

Results shown are directly comparable (except for the shot-peened plates) since all plates were carefully controlled before the treatments were applied, and were identical within normal manufacturing limits. The shot-peened plates were taken from a different production run. The data shown are confirmed by other test curves not included here.

Tests were limited to 50 per cent of the average ultimate strength because in most practical applications higher loads would not be used. Loads approaching one-third of the average ultimate strength are sometimes encountered in unusual chain applications where chain life of a small number of cycles is acceptable. Therefore, the region of the curve shown is that in which most chain loads fall. One specimen of each type was tested to 10 million cycles without failure in order to find the endurance limit. The limits obtained are approximate, further refinement being considered unnecessary at this stage of the research program.

Fig. 10 shows graphically the comparison of endurance limits. While each of the treatments applied to the standard roller link plate shows some improvement, the increase varies in amount from 15 to 75 per cent. Tests on assembled chain would be required to determine how much the fatigue life of assembled chain would be improved by each treatment. However, the tests reported here indicate that roller link plates can be treated to produce residual stresses which increase fatigue life.

DESIGN ABSTRACTS

Synthetic Lubricants

Available in many variations, synthetic oils and greases may be tailor-made to fit special engineering applications

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FOR LACK of a better name, a special class of synthetic oils is often identified as "tailor-made" oils. These are tailor-made in order to obtain certain optimum properties, in the way that synthetic fibers are synthesized to obtain specific qualities. Just as synthetic fibers are not intended to entirely replace cotton or wool, so tailor-made oils are not intended to replace petroleum oils; instead, they are designed for special applications where petroleum products do not function well enough.

Synthetic liquids which have received the greatest attention in the field of lubrication in the past 10 years are the aliphatic esters, the polyalkylene oxides, the silicones, the esters of phosphoric and silicic acids, and the highly fluorine-substituted hydrocarbons. Much progress has been made in relating the molecular structure of such liquids to the viscosity, viscosity-temperature coefficient, volatility, freezing and boiling point, and boundary lubricating properties. A good basis now exists for predicting the properties of many interesting chemical variations of these and analogous materials.

Tailor-made oils have certain inherent advantages over the mineral oils produced by modern methods. Since they are homogeneous in the chemical sense, all of their properties are readily predicted and controlled. Familiar types of chemical addition agents can be used to improve them, and the development of compounded oils is generally simpler, since there are no undesired chemicals present to cause variations from one production batch to another in the response to antioxidants, V. I. (viscosity-index) improvers, detergents, oiliness additives and wear preventives.

When "thick film" or hydrodynamic conditions of lubrication prevail, rheological and heat transfer properties of the oil are dominant. When boundary conditions prevail, significant properties are now known to be not only the physical properties of the surfaces but also the nature of the surface reaction products formed with chemically reactive materials in the oil. There definitely is a contradiction in requiring that a synthetic oil with a low temperature coefficient of viscosity and a low freezing point should not need antiwear additives. Chemists are beginning to recognize that it is much more effective to synthesize oils to provide the necessary antiwear, antiseize and corrosion inhibiting properties. Therefore, in developing either synthetic or petroleum oils, various kinds of chemical additives are nearly always needed to care for the more extreme conditions of lubrication.

Properties: Lubricating oil volatility has become an important property in the past 15 years. This is due to the increasing use of oils of lower viscosity and to the relation of volatility to the rate of oil consumption in some uses and to flammability in others. Thus, in turbojet engines the rate of oil consumption rises with the oil volatility as do the flammability, the rate of lacquering, and the tendency for the oil to boil off the bearing after engine shutdown. Volatility in instrument oils decreases the maximum allowable time before relubrication; in aircraft control bearings it causes the greases to dry up; and in oils and greases for electric motors, it decreases the upper temperature limit of operability and the service lifetime of the lubricant.

Especially useful are oils having the combined properties of low volatility at 210 F, low viscosity at 100 F, and low pour points (or freezing points). In TABLES 1, 2 and 3 will be found data for synthetic oils on:

1. Viscometric properties from 210 to −65 F.
2. ASTM evaporation rate at 210 F.
3. Cleveland open cup flash point.
4. Pour point (or freezing point).

In TABLE 1 are the liquids having viscosities at 210 F of around 10 centistokes; in TABLE 2 are those having viscosities at 100 F of around 10 centistokes; and in TABLE 3 are those having viscosities at −65 F of 1,500 centistokes

Table 1—Properties of Synthetic and Reference Oils

(Viscosities approximately 10 centistokes at 210 F)

Oil	Viscosity (centistokes) −65 F	−40 F	0 F	100 F	210 F	Evaporation rate at 210 F ASTM (% in 22 hr)	Flash point Cleveland Open Cup (deg F)	Pour point (deg F)
Grade 1120 petroleum oil (MIL-0-6082)	.....	....	...	369	24.9	<0.1	545	< 20
Methylphenyl silicone (DC-550)	.....	>60,000	2960	95.4	20.7	<0.1	>600	−62*
Polyethylene oil (SS-903)	.....	....	25,000	219	20.1	...	385	−20
Grade 1060 petroleum oil (MIL-0-6082)	.....	....	...	121	12.4	<0.1	455	< 0
Polyisopropylene oxide diester (Cal. Res. Corp.)	.....	....	...	70.8	12.6	...	465	−55
Polyisopropylene oxide (LB-300-X)	.....	....	4000	65	11.0	0.4	490	−40
Petroleum oil (AN-0-6a)	.....	4000(max)	...	...	10(min)	22(max)	265(min)	<−70
Petroleum hydraulic oil (2-79B)	.....	7500(max)	...	38(min)	10(min)	...	225(min)	<−50
Methyl silicone (DC-200)	300	170	78	22	9.2	<0.1	520	−70
Polyisopropylene oxide (DLB-130 BX)	200,000	13,000	750	28.4	6.5	...	485	<−65
Fluorinated oil (Hooker-Fluorolube Std.)	.....	....	...	86.9	5.6	...	(none)	> 0
Perfluorohydrocarbon (Dupont FCD-332)	.....	....	...	164	4.6	40	(none)	> 0
Pentaerythritol tetracaproate	.....	4690	452	20	4.4	<0.1	477	−40*

* Freezing point.

Table 2—Properties of Synthetic and Reference Oils

(Viscosities approximately 10 centistokes at 100 F)

Oil	Viscosity (centistokes) −65 F	−40 F	0 F	100 F	210 F	Evaporation rate at 210 F ASTM (% in 22 hr)	Flash point Cleveland Open Cup (deg F)	Pour point (deg F)
Bis (2-ethylhexyl) sebacate	8600	1410	187	12.6	3.3	0.1	450	−67
Bis (3, 5, 5-trimethylhexyl) glutarate	17,000	1890	190	11.9	3.2	0.5	...	<−75
Polyisopropylene oxide (DLB-65-B)	17,000	1950	190	11.8	3.2	...	380	<−65
Triamyl tricarballylate	.....	2000	207	10.9	2.6	0.4	...	<−65
Bis (2-ethylhexyl) azelate	8000	1190	156	11.4	3.1	0.2	445	<−75
Synthetic oil (MIL-O-6085)	.....	2000(max)	...	13 ‡	...	1.0(max)	365(min)	−70
Grade 1010 petroleum oil (MIL-0-6081)	40,000	3000(max)	...	10(min)	...	(not required)	265(min)	−70
Dihexyl pinate	6480	1135	149	10.3	2.8	0.3	385	<−75
2-Methyl-1.3 pentanediol dipelargonate	4680	885	123	10.0	2.7	0.5	...	0*
Dipropyleneglycol dipelargonate	.....	....	125	9.5	2.7	1.1	...	−30*
Polyisopropylene oxide diether (Cal. Res. Corp.)	7700	1000	120	9.4	2.9	...	370	<−65
Bis (2-ethylhexyl) pimelate	4970	878	122	9.3	2.7	0.3	...	<−65
Bis (2-ethylhexyl) adipate	4500	807	107	8.2	2.4	0.6	380	<−70
Polyisopropylene oxide (DLB-50 BX)	9500	905	109	8.0	2.5	16.0	350	<−70
Methylsilicone (DC-200)	.....	60	28	7.9	3.5	...	325	−80
1, 5-Pentanediol bis (2-ethylhexanoate)	5600	920	117	7.8	2.2	0.8	...	−50
Perfluorohydrocarbon (Dupont FCD-330)	.....	14,500	297	6.3	1.3	100	(none)	−45
Petroleum hydraulic oil (MIL-0-5606)	.....	500(max)	...	14 §	...	(not required)	200(min)	<−75
Synthetic gas turbine oil (MIL-L-7808)	13,000(max)	....	...	11.0(min)	3.0(min)	(not required)	385(min)	<−75

* Freezing point. ‡ Approximate viscosity at 100 F; specification requires a minimum of 8 centistokes at 130 F. § Approximate viscosity at 100 F; specification requires a minimum of 10 centistokes at 130 F.

Table 3—Properties of Synthetic and Reference Oils

(Viscosities of 1500 centistokes or less at −65 F)

Oil	Viscosity (centistokes) −65 F	−40 F	0 F	100 F	210 F	Evaporation rate at 210 F ASTM (% in 22 hrs)	Flash point Cleveland Open Cup (deg F)	Pour point (deg F)
Special low temperature lubricating oil (MIL-L-17353 BuOrd)	1500 (max)	...	..	5.0 (min)	...	6.0 (max)	300 (min)	<−75
Tetraisooctyl silicate (Cal. Res. Corp.)	1500	365	73	7.6	2.5	2.2	...	....
Bis (2 methylpentyl) glutarate	1390	279	48.1	5.2	1.7	4.1	385	<−65
1,5-Pentanediol (2-ethylbutyrate)	1360	280	48	5.2	1.7	4.9	...	<−65
Dibutyl trimethyladipate	1220	220	43	4.9	1.6	9.1	350	<−65
3-Methyl-1,5 pentanediol bis (2-methyl pentanoate)	924	203	40	4.8	1.7	3.9	355	<−65
Bis (2-methylbutyl) adipate	932	204	39.2	4.8	1.6	5.6	340	<−65
1,5-Pentanediol bis (2-methyl pentanoate)	719	170	35	4.5	1.6	4.8	340	<−65
Bis (amyl*)pimelate	671	180	41	5.4	1.9	2.6	375	<−75
Dibutyl (2-ethyl glutarate)	627	140	28	3.7	1.4	15.1	310	<−75

*"Oxo" process amyl alcohol.

or less. These three groupings are useful because the liquids in TABLE 1 are in the viscometric and boiling point ranges of oils that might be useful at the higher temperatures; those in TABLE 2 are more suited because of viscometric and low pour point properties for use in equipment exposed to a wide range of temperatures; while those in TABLE 3 are especially suited to applications in fine instruments or other mechanisms having low power input for operation at ordinary and extremely low temperatures.

Although nearly all of the liquids in TABLE 1 have flash points well above 400 F and evaporation rates of under 0.1 per cent, few have pour points much below 0 F. The fluorocarbons are noteworthy because they have pour points above 0 F and evaporation rates which are much greater than those of any other oils in this group. Polyethylene oil has one of the lowest flash points of all and a pour point of only −20 F. But silicones, polyalkylene oxides and high molecular weight aliphatic esters are each becoming increasingly more useful as high temperature oils because of their low volatilities, high flash points, and nonsludging characteristics.

A wide spread in the evaporation rates is exhibited by the liquids listed in TABLE 2 especially when the low pour-point petroleum oils are compared with most of the synthetics. Many esters, doubly chain-stoppered polyalkylene oxides, and methyl silicones have evaporation ranges of from 0.1 to 0.5 per cent as compared to 8 to 20 per cent or more for the petroleum products. It will be noted that many of the esters have lower evaporation rates and higher flash points than the other liquids. Viscosities at −65 F range from 5,000 centistokes in some of the synthetics to 20,000 in others, while the petroleum products have values of 40,000 centistokes or more. The extremely high volatility of fluorinated oil FCD-330 is characteristic of the perfluorohydrocarbons having low pour points.

When liquids are required in the viscosity range of TABLE 3, there are obvious advantages in employing synthetic oils. Thus, grade 1005 petroleum oil, which has a viscosity at 100 F comparable to most of the liquids in TABLE 3, has a viscosity at −65 F of around 2,000 centistokes, a flash point of 225 F or less, and an evaporation rate of 20 per cent. In marked contrast, most of the synthetics in this group have viscosities at −65 F of 700 to 1,500 centistokes, flash points of 340 to 385 F and evaporation rates of 3 to 6 per cent.

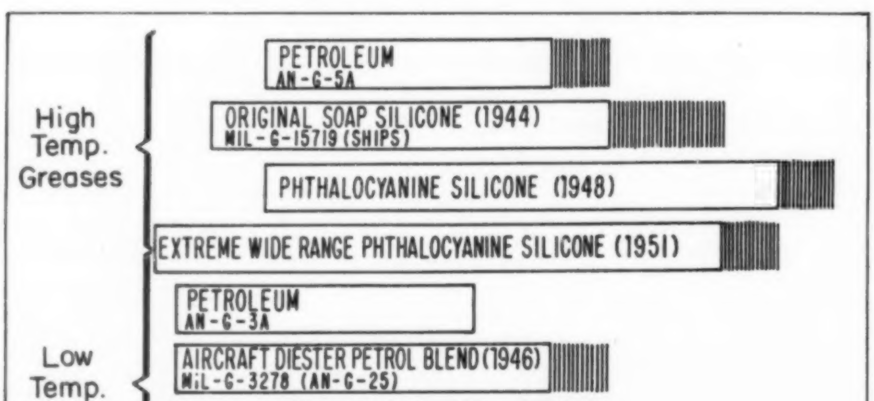
Many of the other properties of these synthetic oils are now fully described in the literature and so will only be mentioned briefly. These include the limited oiliness properties of silicones and their excellent oxidation resistance; the stability and nonflammability of the fluorocarbons as well as the limited solubility for oils and additives; the appreciable detergency of ester oils, their excellent response to inhibitors and oiliness additives, their freedom from engine deposits and sludging, their compatability with petroleum oils and polyisopropylene oxides, and their deleterious effects on certain types of plastics, paints, and rubber; and finally, the detergency of polyalkylene oxides, their freedom from sludging and lacquering, their peculiar water solubility properties, the problem of obtaining a high degree of rust inhibition in some applications, and their attack on some types of paints.

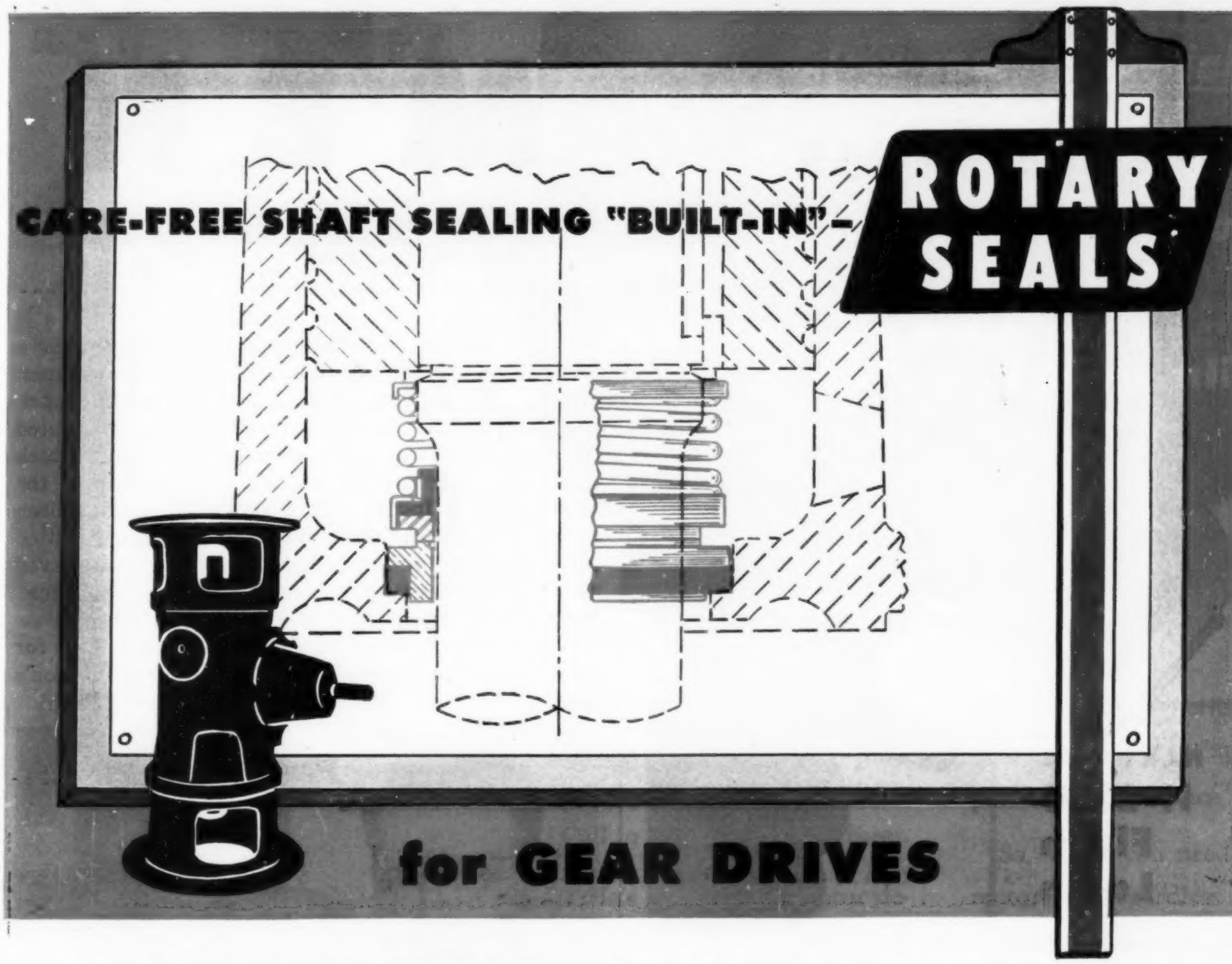
In comparing or classifying the synthetic oils, it should be recognized that, as with synthetic fibers and plastics, no one oil is best for all uses. Much research has been directed to learning the proper fields of application of each of these new liquids, how to blend them, and how to modify their properties with chemical addition agents. Naturally, engineers would like to know which synthetic oil is best for each type of application or else in which oil there exists each important property to the highest degree. However, the frequent existence of physical, chemical and economic requirements peculiar to each lubrication problem can make it difficult to use either of these two types of general comparison. Wherever possible each potential user or equipment designer should seek guidance from specialists about the applicability of the new synthetic lubricants to his own problems.

Greases: New greases made from synthetic oils were introduced early in World War II commencing with the aliphatic esters and followed by the polyalkylene oxides, the silicones, the silicone-diester blends, and more recently, the fluorinated hydrocarbons. In the postwar years there has been increasing interest

(*Continued on Page* 255)

Fig. 1—Extreme temperature synthetic greases. Cross hatch areas indicate temperature in which lubricant functions, but with shortened life





Gear Drives are used in all kinds of rugged applications, both here and abroad. Units for such purposes as dry dock installations, for example, must be designed for day in and day out performance, without the frequent down-time for repairs often encountered with old-fashioned Shaft Sealing methods. That's why you'll find ROTARY SEALS used throughout major lines of gear drives—to provide *Certainty* in that critical factor of Shaft Sealing.

Most ROTARY SEALS are specially adapted for a given application from the basic ROTARY SEAL principle. Our engineers, experienced in the solution of Shaft Sealing problems for manufacturers large and small in many widely varying fields, will be glad to consult with you on *your* requirements. It's wise to call us in at the drawing board stage—we can often suggest the simplest design approach from the Shaft Sealing Standpoint, based on successful jobs in many lines.

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is the original approach to a practical solution of a universally troublesome problem. Our booklet "SEALING WITH CERTAINTY" explains and illustrates the principle. We're glad to send it to you without obligation.

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MEN OF MACHINES

Dorian Shainin and **Franklin E. Satterthwaite** have joined Rath & Strong Inc., industrial consultants, Boston. Mr. Shainin, who has done outstanding work in developing new quality control methods, originated the "Lot Plot" system of sampling inspection which has found wide use in industry. A graduate of the Massachusetts Institute of Technology, he has been associated with the Hamilton Standard Div. of United Aircraft Corp. for the past 16 years. He has served as seminar chairman at meetings of the American Management Association, is a fellow and former New England regional director of the American Society for Quality Control and chairman of the association's

Dorian Shainin

F. E. Satterthwaite

Aircraft Technical Committee, and is past chairman of the Eastern Region Inspection Committee of the Aircraft Industries Association. He won the ASQC 1952 Brumbaugh Award for the paper that has made the greatest contribution to the development of industrial application of quality control. Mr. Shainin, who has written other articles for MACHINE DESIGN, is the author of the current series "Quality Control Methods," Part 7 of which begins on Page 150 of this issue.

Dr. Satterthwaite received his doctorate in mathematics and statistics at the University of Iowa. For the past six years he has served as quality control engineer and statistical consultant for General Electric Co. in its appliance, wire, small motor, transformer and chemical operations. He is an authority on the application of statistical methods to research and engineering and has developed quality control procedures through cost analysis and incorporated all avail-

able information, statistical and nonstatistical, into quality control decisions. Dr. Satterthwaite is a director and fellow of the American Society for Quality Control and a member of its Educational Committee. He is also a member of the American Chemical Society, the American Institute of Electrical Engineers, Society for the Advancement of Management, American Society for Engineering Education, Institute of Mathematical Statistics and American Mathematical Society.

♦

Thomas J. Kelly has been appointed manager of engineering for the General Electric Co. Appliance Control department at Morrison, Ill. He will be in charge of development and design of the department's products.

♦

The newly created position of assistant director of engineering of the Tractor Div. of Allis-Chalmers Mfg. Co., Milwaukee, has been filled by **W. L. Voegeli,** who has been associated with the company since 1935.

The company also recently announced that **Gerald E. Smart,** a plant engineer at the Norwood, O., works since January 1951, has been placed in charge of the plant engineering department there. He had been engaged in tool engineering work at the West Allis works before transferring to Norwood.

♦

Bernard James Rainey, recently named to the engineering staff of Auburn Button Works Inc., Auburn, N. Y., has been assigned to the company's vacuum forming division. He was formerly a design engineer for John Verdiun Machine Corp., Paterson, N. J.

♦

Ryan Aeronautical Co., San Diego, Calif., has announced the appointment of **G. C. Danch** and **A. Deyarmond** to key posts in its engineering department. Mr. Danch, who has been associated with the development of fighter planes at the Fighter Branch of the Navy Bureau of Aeronautics, will serve as executive assistant to the director of engineering and will perform special assignments. Mr. Deyarmond was chief of structures for Ryan from December 1946 to April 1948; prior to his present appointment he has been chief of the special studies office of the Air Technical Intelligence Center at Wright-Patterson Air Force Base. He will again serve as chief of structures and will also be chief of aerodynamics.

♦

Westinghouse Electric Corp., Pittsburgh, recently announced several new appointments in two of its divisions. **C. B. Campbell** was named consulting engineer, and **J. R. Carlson,** manager of engineering for the Steam Div. Mr. Campbell joined Westinghouse in 1919 as a technical apprentice, has held various supervisory positions in the division's engineering department, and since 1944 has been manager of engineering. In 1926 Mr. Carlson also joined the company as a technical apprentice. He has been associated

with the engineering department since 1941 and has served as manager of the general engineering section since 1947.

Parker W. MacCarthy is now manager of the jet engine section, electric appliance engineering department, at the company's Electric Appliance Div. plant in Columbus, O. Formerly in charge of compressor design for domestic refrigeration units at the division's East Springfield, Mass., plant, he will now be responsible for transmitting design information to the factory for the manufacture of jet engine components.

♦

To direct production engineering activities, **F. Clark Cahill** was recently appointed chief engineer of the engineering and production division of Airborne Instruments Laboratory Inc., Mineola, L. I., N. Y. He formerly served as supervisor of the radar section in the laboratory's research and engineering division.

♦

Greer Hydraulics Inc., Brooklyn, N. Y., has appointed **F. W. Gottschling** as administrative engineer. He will assist the president and chief engineer. Mr. Gottschling will supervise the scheduling of all engineering projects and the utilization of engineering manpower and will also co-ordinate the engineering department functions with those of other departments of the company.

♦

William T. Stephens has joined the Parker Appliance Co., Cleveland, as staff engineer in charge of the company's development program of industrial hydraulic equipment. He was formerly vice president and general manager of the Valve Service Co., Willoughby, O. Prior to that association he had been affiliated with the Commercial Shearing and Stamping Co., where he rose to the position of chief engineer and manager of the hydraulic division, and also helped establish Hydreco Inc. in 1935, served as chief engineer of this concern for ten years and then acted as a general hydraulic consultant before joining Valve Service Co.

♦

Loewy Construction Co. Inc., subsidiary of Hydropress Inc., New York, has announced the promotion of **H. Albers** to chief engineer in charge of the United States Air Force Heavy Press Program, and the appointment of **G. Krause** as chief designer in charge of the same program.

♦

Key personnel for the government's huge atomic energy plant which is now under construction near Portsmouth, O., is being organized at offices of the newly created Goodyear Atomic Corp., Akron, O., subsidiary of The Goodyear Tire & Rubber Co. **William A. Brown** heads the staff engineering and maintenance division, and **Hugh F. Porter Jr.** is head of the electrical and instrument division. **Thomas W. Leary** has been appointed superintendent of process maintenance; **Arthur J. Brust,** superintendent of planning

and scheduling; and **Nathan H. Hurt Jr.**, superintendent of staff engineering. These men were all transferred from other posts in the Goodyear organization.

Two newly appointed development engineers are **William H. Taylor,** who will serve as superintendent of design and development, and **Howard L. Caterson,** who will be head of inspection.

♦

Pioneer researcher in electronics, including television, **V. K. Zworykin** has been awarded the 1952 Edison Medal by the American Institute of Electrical Engineers "for outstanding contribution to the concept and design of electronic components and systems." Dr. Zworykin is vice president and technical consultant of the RCA Laboratories Div., Radio Corp. of America, Princeton, N. J. He has been associated with RCA since 1929. The medal will be presented at the opening session of the Winter General Meeting of AIEE in New York on January 19.

♦

Edwin Crankshaw has been named chief engineer and head of the engineering division of The Cleveland Graphite Bronze Co., largest operating unit of the newly formed Clevite Corp., Cleveland. He succeeds **Henry W. Luetkemeyer,** who is now one of the key men in the new products division of Clevite. Mr. Crankshaw came to Cleveland Graphite as product design engineer in 1942.

♦

Formerly chief engineer at Link Aviation Inc., Binghamton, N. Y., **W. W. Wood** has been appointed vice president in charge of engineering.

♦

Recent appointments in the Instrument Div. of Allen B. Du Mont Laboratories Inc. are those of **H. B. Steinhauser** as manufacturing engineer, **L. E. Florant** as head of the engineering services section, and **A. W. Russell** as head of the electrical design section. Mr. Steinhauser, formerly a senior engineer, has been engaged in product design for the division since 1950. Mr. Florant was previously an intermediate engineer for the division, and Mr. Russell was a senior engineer.

♦

Stackpole Carbon Co., St. Marys, Pa., has appointed **E. I. Shobert** as manager of carbon research and engineering and **Henry M. Dressel** as director of research and engineering for the electronic components division. **L. D. Andrews** is now director of research and engineering on magnetic materials, **E. F. Kiefer** has been appointed director of research and engineering on carbon products, and **F. X. Sorg** is director of research and engineering on fixed resistors.

♦

Three new staff members of Arthur D. Little Inc. Cambridge, Mass., are **Garth A. Abbott,** who has joined the engineering physics and special devices group, and **Jay Joseph Martin Jr.** and **Donald Guild,** who are new members of the mechanical division.

ANNOUNCING!

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National, a world leader in oil seals, now offers a complete line of O-Rings

You know National's reputation as a leading manufacturer of oil seals. Now, you can expect the same quality of product, the same good service on O-Rings. With the new distribution system now being set up, National O-Rings will be offered through a wide-spread network of applications engineers and distributors.

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NEW

THE ENGINEER'S
Library

Recent Books

Electronic Analog Computers. *By Granino A. Korn and Theresa M. Korn; 394 pages, 6 x 9 inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, $7.00 *postpaid.*

Electronic computing devices are becoming increasingly important as a means of surmounting praccal difficulties, such as necessary time, funds and skilled personnel, involved in the solution of engineering problems requiring complex numerical calculations. In this book, attention is centered on the dc analog computer which has been employed with notable success in the design of automatic control systems. An elementary knowledge of electronics and differential equations is required for understanding. The techniques of computer design and application described are those which have become well established during the last five years. Particular emphasis is placed on determination of scale factors, which experience has shown to be the most prevalent source of error.

An introductory chapter briefly sketches the overall scope of computer operation and is followed by a chapter which details a simple standard procedure for setting up a dc analog computer to solve differential equations. Succeeding chapters cover computer setups for several representative practical problems; design of the various electronic computing elements for maximum accuracy; developments in analog multiplication and function generation; design of auxiliary components; and operating and checking procedures. In the concluding chapter, design and construction of complete installations for industrial applications are discussed. Also included is an appendix which lists properties of parallel feedback-type operational amplifiers.

•

Metallurgy for Engineers. *By John Wulff, Howard F. Taylor and Amos J. Shaler, department of metallurgy, Massachusetts Institute of Technology; 632 pages, 5½ by 8½ inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, $6.75 *postpaid.*

Although written primarily for the engineering student, this textbook has also been designed as a reference for the practicing engineer. Extractive metallurgy has been omitted and the mathematical considerations of metal working have been simplified. Emphasis has been placed on those fundamentals necessary for an adequate understanding of metals,

their selection and use. The first half of the book discusses the concepts and principles underlying metal processing; the second half deals with the processes themselves. Brief chapter summaries and reference lists are provided at the end of each chapter.

♦

Methods of Applied Mathematics. *By F. B. Hildebrand, associate professor of mathematics, Massachusetts Institute of Technology;* 536 *pages,* 5½ *by* 8¼ *inches, clothbound; published by Prentice-Hall Inc., New York; available from* MACHINE DESIGN, $7.75 *postpaid.*

Fundamental facts and techniques relevant to four fields of advanced mathematics provide the subject matter for this textbook. The topics covered are those usually not considered under the general classification of advanced calculus. Material has been organized to permit self-study; interdependence among the various chapters has been minimized as much as possible.

In the first chapter, matrices, determinants and linear equations are discussed. The second chapter takes up the calculus of variations and applications and is followed by a chapter on difference equations. Concluding is a chapter which deals with integral equations. An appendix containing the Crout method for solving sets of linear algebraic equations has also been included.

♦

Fractional H.-P. Electric Motors. *By F. G. Spreadbury;* 352 *pages,* 5½ *by* 8½ *inches, clothbound; published by Sir Isaac Pitman & Sons Ltd., London; available from* MACHINE DESIGN, $7.50 *postpaid.*

In principle, fractional horsepower motors are no different from their integral counterparts; however, certain differences, such as the number of poles, power losses and types employed, set the smaller motor apart as a separate entity. This British book deals with the principles, performance, control and design of small motors. Its aim is to supplement the present literature on integral motors which is not always applicable to the fractional types.

Main portion of the text material is devoted to the various types of motors. Direct-current, polyphase induction, single-phase induction, universal, repulsion, synchronous and other miscellaneous motors are discussed in turn. Concluding chapters cover performance characteristics and motor control.

♦

Modern Plastics Encyclopedia and Engineer's Handbook. 848 *pages,* 8¼ *by* 11¼ *inches, clothbound; available from Plastics Catalogue Corp.,* 575 *Madison Ave., New York* 22, *N. Y.,* $2.00.

Issued annually since 1941, this latest edition (1952) describes the commercially available plastics materials with emphasis on their suitability for spe-

cific end uses. It presents a picture of technical advances in the plastics industry during the past twelve months.

♦

Design and Use of Cutting Tools. *By Leo J. St. Clair; 451 pages, 6 by 9 inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, $7.00 *postpaid.*

Basic considerations in the selection, design, grinding and usage of machine cutting tools are discussed.

Association Publications

Fifty-Year Index to ASTM Technical Papers and Reports. 215 *pages,* 6 *by* 9 *inches, clothbound; available from American Society for Testing Materials,* 1916 *Race St., Philadelphia* 3, *Pa.,* $6.00 *per copy.*

This index provides detailed author and subject listings for the technical papers and reports dealing with materials, particularly their properties and testing, appearing in ASTM publications during the period of 1898 through 1950.

Government Publications

Investment Precision Casting. *Report PB* 111 001 *by Roy W. Tindula;* 13 *pages,* 8 *by* 10½ *inches, paperbound; available from Office of Technical Services, U. S. Dept. of Commerce, Washington* 25, *D. C.;* 25 *cents.*

History, uses, process details, precision and limitations of the investment or "lost-wax" casting process are discussed in this report prepared by a research metallurgist of the OTS. Included is an extensive bibliography which lists references to periodicals, government reports and books on the process.

♦

NACA Technical Series. *Each publication is* 8 *by* 10½ *inches, paperbound, side-stapled; copies available from National Advisory Committee for Aeronautics,* 1924 *F St. N.W., Washington* 25, *D. C.*

The following Technical Notes are available:

2760. **Derivation of Stability Criterions for Box Beams with Longitudinally Stiffened Covers Connected by Posts**—21 pages

2769. **Experimental and Theoretical Determination of Thermal Stresses in a Flat Plate**—35 pages

2771. **Thermal Buckling of Plates**—39 pages

2778. **Stress and Strain at Onset of Crazing of Polymethyl Methacrylate at Various Temperatures**—21 pages

2779. **Effects of Moderate Biaxial Stretch-Forming on Tensile and Crazing Properties of Acrylic Plastic Glazing**—42 pages

2782. **Bending of Thin Plates with Compound Curvature**—49 pages

2788. **Effects of Solvents in Improving Boundary Lubrication of Steel by Silicones**—23 pages

2791. **Correlation of Tensile Strength, Tensile Ductility, and Notch Tensile Strength with the Strength of Rotating Disks of Several Designs in the Range of Low and Intermediate Ductility**—30 pages

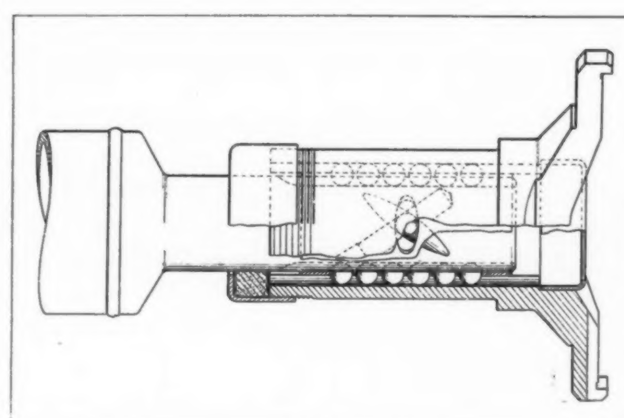
2798. **An Experimental Investigation of the Behavior of 24S-T4 Aluminum Alloy Subjected to Repeated Stresses of Constant and Varying Amplitudes**—23 pages

NOTEWORTHY

Patents

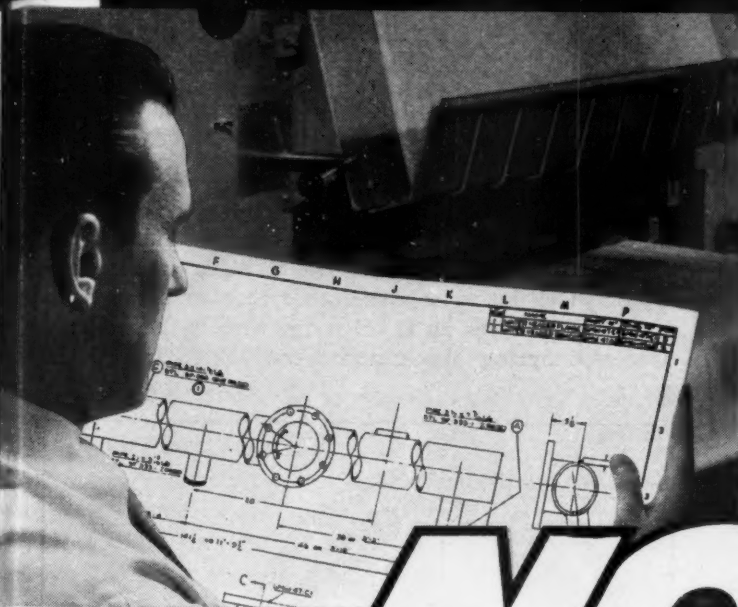
NOISELESS PRESSURE CONTROL for high-pressure hydraulic systems is accomplished with the valve covered in patent 2,600,875. Excessive system pressure actuates a cylindrical spring-loaded plunger which opens into a low pressure chamber to provide relief. Design of the plunger provides a slow opening and closing action, obviating the chattering, vibration or squealing usually associated with this type of valve. Unique features include a built-in integral dashpot, double-sealing of the plunger, full cushioning during opening and closing, and a safety arrangement which permits pressure relief even though the spring may fail. The patent has been assigned to Hydraulic Equipment Co. by Edward Hrdlicka.

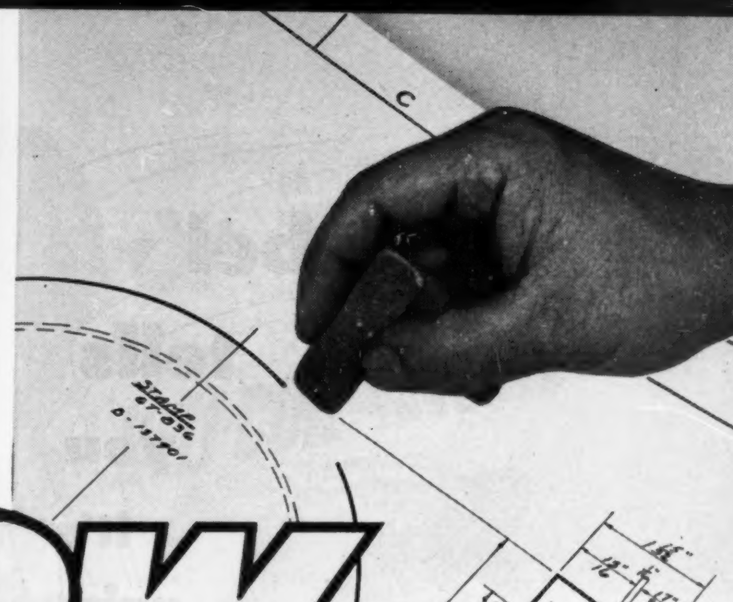
MULTIPLE BALL KEYS eliminate sliding friction in a telescopic shaft coupling and thereby afford substantially longer life with less backlash than is experienced with sliding key joints. Additionally, rolling action of the driving balls assures smooth telescopic action of the joint under full torque. Uniformly



spaced by a tubular cage, the two rows of driving balls are limited in their axial movement through a master ball carried in a circumferential slot of the cage. This ball also engages diagonal grooves in the shaft and sleeve members of the joint and prevents telescopic action beyond predetermined limits. Patent 2,605,622 assigned to Borg-Warner Corp. by Edmund B. Anderson.

TORSIONAL OSCILLATION DAMPING is effected in a built-up power transmission sprocket by a series of rubber-metal sandwich members which serve





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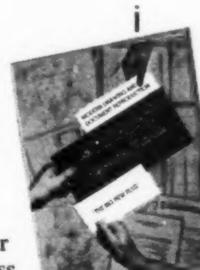

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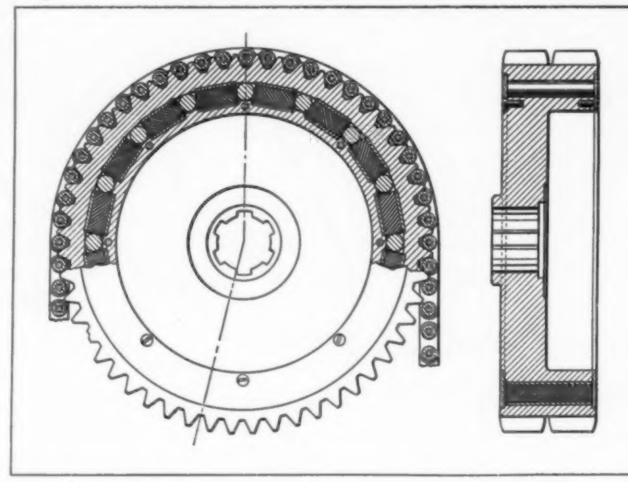
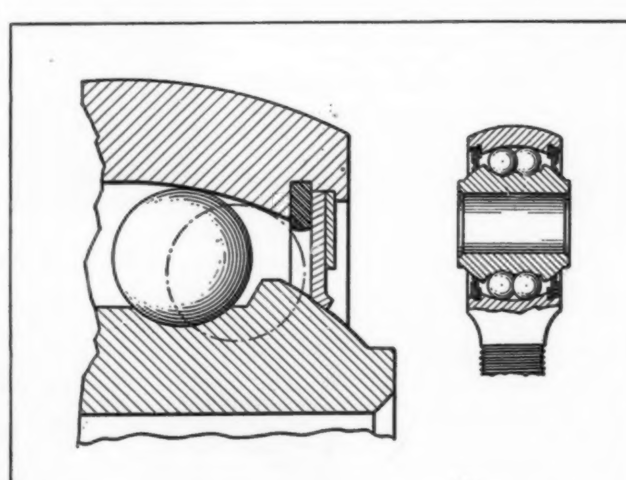
as keys between the separate rim and hub of the sprocket. Tendency of the keys to shear under pulsing or shock loads isolates rather than amplifies such undesirable forces as is often the case with similarly used coil spring shock absorbers. Concentricity of



the rim and hub is maintained through steel antifriction rollers which engage circular surfaces inside the rim and outside the hub between each shock absorber. Circular side plates retain the rollers and keys, and maintain axial position of the rim. Patent 2,596,501 assigned to the New Britain Machine Co. by Donald H. Montgomery.

GREASE SHIELD PUSHOUT in extreme angles of displacement is prevented in a self-aligning ball bearing rod end by bumper rings snapped into grooves inside the shield elements. Ball contact with the inside corner of the rings limits the angularity of the races,



so that ball thrust against the seal washers and their retaining rings is avoided in all positions. This construction also enables removing the grease shields for

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they save money because they strengthen threads and make *fewer* fastenings do the same holding job...they make *lighter* bosses and flanges practical...and they *save* weight in *two* ways: (1) by permitting use of cap screws instead of bolts and nuts or studs and nuts; (2) by putting smaller, shorter, fewer cap screws to work than required for unprotected threads.

they protect your product from thread wear and stripping for life in every kind of metal, in plastics or wood. They preserve your customer good-will by preventing complaints about thread failure.

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Heli-Coil Inserts are precision formed thread liners of stainless steel or phosphor bronze wire, diamond-shaped in cross-section. They are available for National Coarse, National Fine and Unified threads, pipe threads and aviation and automotive spark plug threads. They are made in all standard sizes and lengths for assemblies requiring Class 3, 3B, 2 or 2B fits.

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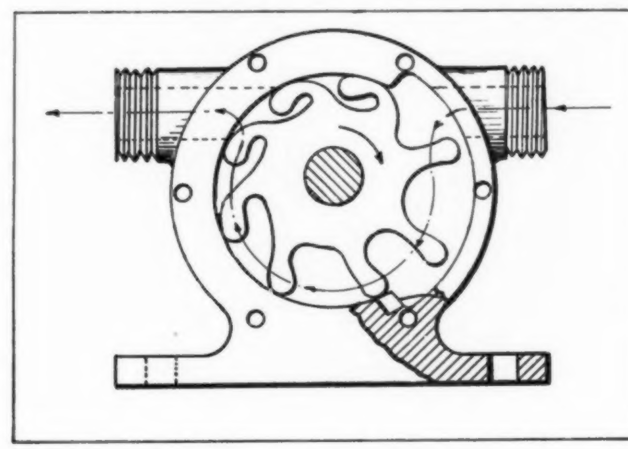
Noteworthy Patents

bearing inspection or relubrication without danger of the balls spilling out. Patent 2,584,227 assigned to the Fafnir Bearing Co. by Howell L. Potter.

STRESS-FREEING OSCILLATION of resilient rotary pump vanes following the discharge cycle prevents permanent set in the vane material. The pump, covered in patent 2,599,600 assigned to the Cascade Pump Co. by Albert W. Arnold, utilizes a flexible



molded rotor offset to provide pumping action. Deflection of the vanes produces the initial suction as well as the pressure necessary for discharge. After discharge the vanes pass into an enlarged region which permits them to spring out and dissipate the energy absorbed through deformation.

ONE-WAY CLUTCHING for small mechanisms is achieved with a single leaf spring in patent 2,601,911. Invented by Zoltan Takats and assigned to the General Aniline & Film Corp., the clutch has been designed to meet the needs of motion picture film-feeding devices. An S-shaped flat spring is used as the power transmission element; engagement of the spring ends with the inner surface of a cylindrical member provides a positive one-way drive and at the same time permits overrunning.

FRICTION-RETAINED NOSEPIECE is utilized as the face sealing element in a diaphragm-type face seal described in patent 2,601,996. Designed for use between two rotating members, the seal can accommodate repeated end-play or angular movements. The nosepiece, retained in a tapered recess at one end of the diaphragm, facilitates assembly and replacement. Several construction modifications of the seal for different types of installations are shown in the patent which has been assigned to Chicago Rawhide Mfg. Co. by John H. Sefren.

Engineering News Roundup

Instrument Records Speed Of Landing Airplane

Eliminating the need for recording on film, which is more expensive and necessitates a delay in obtaining results, an electrical-optical "brain" gives instant data on how fast an airplane lands on the deck of a carrier. This Touchdown Rate of Descent Indicator, nicknamed TRODI, is being produced by North American Aviation Inc.

In operation, TRODI sends out two parallel beams of light which are thin vertically, wide horizontally, and one foot apart. A mirror system on the incoming airplane cuts the top beam, reflecting the light back to a photoelectric cell, which starts an electrical charge into a condenser. The descending airplane then cuts the second, lower beam, reflects it, and stops the charge going into the condenser. The electrical charge stored during the interval between breaking the two beams is quickly translated from voltage to rate of descent in feet per second. Measurements are accurate to 0.4-foot per second.

All new Navy airplanes must be tested for rate of descent before they are placed in regular operation. Formerly it might be several days before a pilot would know if he had come in at the correct rate of descent; this new instrument has the data ready for the pilot immediately.

Besides measuring the rate of descent that aircraft structures can withstand, the instrument helps evaluate landing characteristics of airplanes controlled by either automatic or manual systems, helps instructors teach student pilots to land, and aids in determining landing conditions suitable for various aircraft.

Milford Proposes Method For Analyzing Fasteners

What must be considered in a new design involving the use of rivets or tubular fasteners? Milford Rivet & Machine Co. has proposed an organized procedure for analyzing this problem for cold-headed fasteners.

Such fasteners may be one of three different types: (1) standard tubular rivets, which are simplest to make and lowest in cost, (2) modified standards, which require variation from standard dimensions and sometimes include secondary operations, and (3) cold-headed specialities which require special designs and tools to produce.

An attempt should first be made to see if a standard rivet will serve the purpose. If not, then one of the other two types must be used. At the same time, ease and economy in setting the fastener in place must be considered. Proper application of automatic rivet-setting machines for standard and special tubular split rivets is a prerequisite for economical assembly. Specialized fasteners, if properly designed, can often be applied with automatic machines.

Cold-headed specialties, while more expensive, offer certain advantages, among which are strength and toughness, favorable grain positioning, and different surface finish if desired.

New Exposition To Feature Basic Materials in Industry

The "Exposition of Basic Materials for Industry," a new show and conference which will cover a whole range of new materials, their properties and potentialities, is scheduled for June 15 to 19 at Grand Central Palace, New York. Top executives of 20 major companies are included in the sponsoring board.

Simultaneous with the exposition, a series of technical conferences will be conducted to discuss new materials and their potentialities. The event is intended to create a clearing house of information and ideas for engineers and executives by offering an integrated picture of the vast and complicated structure of materials fundamental to product development.

Provisions for an attendance of at least 15,000 are being made. Almost 2000 experts will be on hand

to lead conference discussions and answer questions.

"New materials are being developed almost daily," said Don G. Michell, president of Sylvania Electric Products Inc. and chairman of the board of sponsors," . . . it is not only consumer products which are undergoing change. Jet propulsion, atomic power and huge increases in the speed of mechanical production are presenting problems in industrial materials almost undreamed of in the thirties."

Admission will be open for management executives, project and design and production engineers. Clapp & Poliak Inc. will manage the exposition.

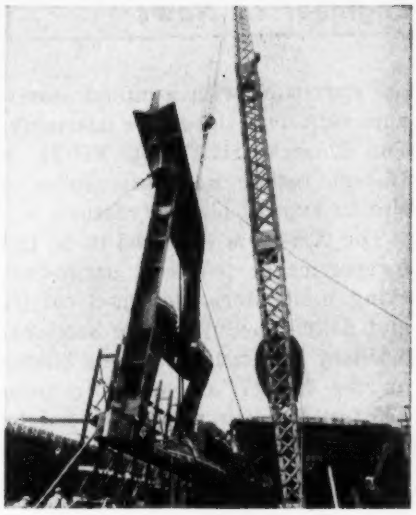
Develop New High-Strength Alloy

Composed of copper, nickel, silicon and a small amount of iron, Strenicor, a new alloy developed by General Electric in co-operation with Revere Copper and Brass Inc., is not susceptible to stress corrosion or season cracking when clamped to any tension within the ultimate strength of the material. When made by sand casting, the alloy has a tensile strength of 90,000 psi and a yield strength of 70,000 pounds; made by hot forging, it has a tensile strength of 107,000 psi and a yield strength of 83,000 pounds.

Supplies Aircraft Carrier Propulsion Equipment

Consisting of four cross-compound turbines and four double-reduction gears, propulsion equipment supplied by the General Electric Co. for the new U. S. Navy aircraft carrier CVA-60 will be the most powerful ever to be used on a warship.

Economy of operation resulting from higher temperatures and pressures in the steam turbines will enable the carrier to travel approximately 20 per cent farther with the same amount of fuel than it could with the type of equipment used in World War II ships. Now under construction, the new carrier is scheduled for completion in 1955.



One of the largest steel casting assemblies ever produced for this type of ship construction, a 48-ton cast structural unit serves as the framework which supports the wheel shaft, propeller and rudder of a new 647-foot ore carrier built for service on the Great Lakes. The casting measures 28 feet in height and 25 feet in length. Because of its size, the unit was cast in five sections which were then welded together. The sections were cast by Penn Steel Castings Co.

•

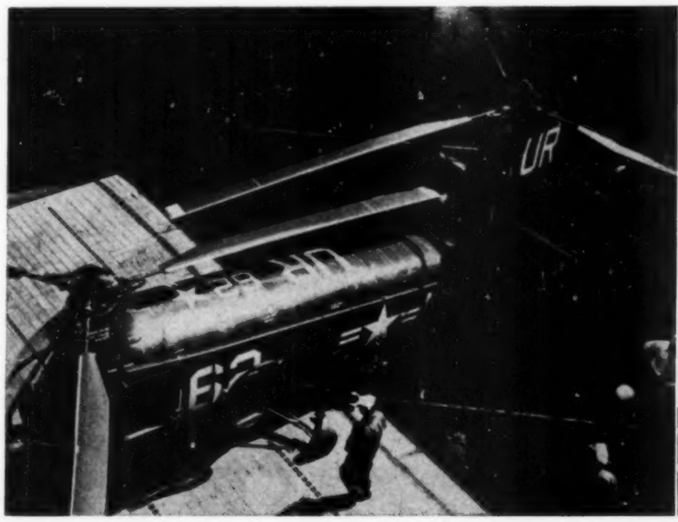
Giant Helicopters Show Ingenious Design

Two new helicopters, both in the heavy lift class, show increased use of newer design principles. The first, Piasecki's new 44-place XH-16 helicopter, uses compression-formed spar tubes which form the backbone of the honeycomb-structure rotor blades. The XH-17, an experimental heavy lift cargo carrier which has just made its first flight, is powered by two modified turbojets.

Spar tubes in the XH-16 rotor blades are extruded tubes compression formed and cold forged by the Rockrite process by Tube Reducing Corp. These spar tubes carry the full load placed on the rotor blades, consisting of the entire helicopter weight.

Rotor blade assembly consists of a honeycomb structure mounted on

Piasecki HUP helicopter, left, uses compression-formed Rockrite spar tubes as a "backbone" for the honeycomb-structure rotor blades. A similar type of construction is also used on the YH-21, right, a 16-seat "work horse" used as a rescue helicopter by the Air Force




the spar tube with a milled aluminum skin over the entire assembly. The Piasecki HUP and YH-21, a 16-seat rescue helicopter, uses a similar type of blade structure.

The XH-17 is expected to be the forerunner of powerful cargo-carrying helicopters, designed to lift and deliver heavy bridge sections, artillery and trucks. Rotor blades on the XH-17 extend more than 125 feet from tip to tip, and height of the machine is more than 30 feet.

Production design department in the new building of Sundberg-Ferar has individual "studies" or compartments for staff designers. According to Carl W. Sundberg and Montgomery Ferar, staff designers working in their new Royal Oak, Mich., headquarters "are provided with a psychological lift by working in a creative atmosphere which includes privacy, good lighting, sound-proofing and air conditioning. Facilities for model making provide a completely integrated operation whereby designers can supervise the development of product ideas into three-dimensional models."

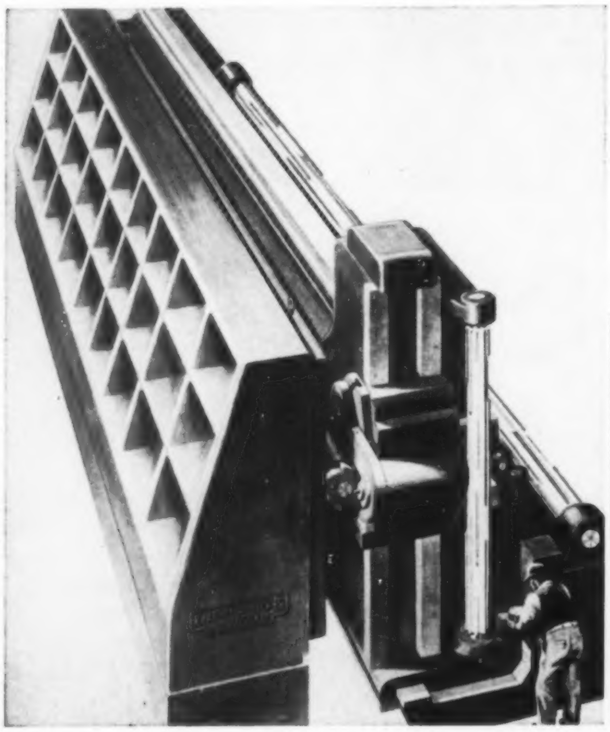
Huge Special Machines Built in New Plant

Building of huge special machine tools weighing as much as 250 tons is one of the activities contemplated for the new Kearney & Trecker Corp. plant (MACHINE DESIGN, December, Page 260).

The machine tools will be built mainly for machining die blocks, dies and parts forged on new heavy forging presses now being installed for making aircraft parts. Presses up to 50,000 tons are currently being built and installed as the result of government orders, involving the use of die blocks weighing upwards of 10 tons. Gigantic aircraft components forged on these presses would be 15 to 20 feet long and about one-half as wide.

Although the defense program will have first call on the production of the new plant, a great share will be available for nondefense production, according to Francis J. Trecker, president. Equipment probably will be fairly complicated, involving the use of electronic controls, photoelectric devices and three-dimensional tracer mechanisms on these machines.

Cost of the new plant will be approximately $5.2 million. More than $2.5 million worth of machine tools will be installed, including some 60 large units. Total space of 193,000 square feet will be available, of which 173,000 square feet will be manufacturing space. About 50 design engineers will be needed for the new plant.



Left—Projected design for large bed-type milling machine for milling integrally-ribbed aircraft skins. One of the proposed machines to be built at a new Kearney & Trecker plant, the machine would be 50 feet long, 18 feet high and 27 feet wide. Approximate weight would be 120 tons and table width, 9 feet

Below—Vertical style skin mill, another projected design

Farval saves time in lubrication of Electroweld Tube Mills

FORMING steel tubing at high non-stop speed was a prime consideration to the builder of this electroweld tube mill. Slow, old-fashioned hand oiling just couldn't lubricate 44 main bearings properly and still keep step with accelerated production schedules.

No wonder this manufacturer installs Farval Centralized Lubrication Systems on his fast-moving continuous mills! Farval lubricates *while the mills are in full operation,* entirely eliminating downtime for oiling and protecting the costly bearings which help keep exactly to specification the 15,000 feet of tubing put out each 8 hours.

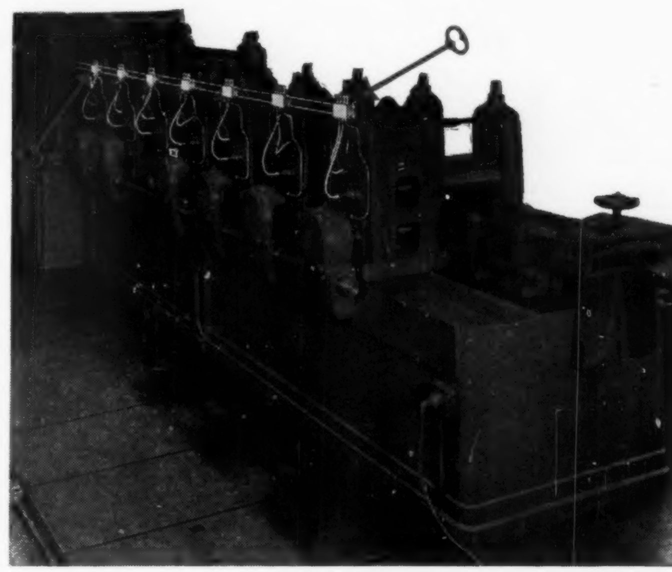
Today Farval systems safeguard millions of bearings in steel and metal working plants—on sheet and plate mills, tube and pipe mills, blooming mills—in fact, every type of equipment that must be lubricated regularly and adequately, to keep bearings on the job no matter how hard to get at.

Farval—proved in 25 years' service—is the original Dualine system of centralized lubrication that delivers oil or grease under pressure to a group of bearings from one central station, in exact quantities, as often as desired. The Farval valve has only two moving parts—is simple, sure and foolproof, without springs, ball-checks or pinhole ports to cause trouble. Indicators at every bearing show that each valve has functioned.

For a full description of time-saving Farval, write now for Bulletin 25. The Farval Corporation, 3265 East 80th Street, Cleveland 4, Ohio.

Affiliate of The Cleveland Worm & Gear Company, Industrial Worm Gearing. In Canada: Peacock Brothers Limited.

FARVAL — *Studies in Centralized Lubrication*
No. 127



KEYS TO ADEQUATE LUBRICATION—*Wherever you see the sign of Farval—the familiar valve manifolds, dual lubricant lines and central pumping stations—you know that a machine will be properly lubricated. Farval manually operated and automatic systems protect millions of industrial bearings.*

Photo by courtesy of American Electric Fusion Corporation, builder of this tube mill.

FARVAL

CENTRALIZED SYSTEMS OF LUBRICATION

Engineering News

Builds Supersonic Delta-Wing Interceptor

First supersonic delta-wing interceptor to be built for the U. S. Air Force, the Convair F-102 is designed for "very high speeds in the stratosphere and incorporates significant improvements in electronics and armament," according to company president J. T. McNarney. Details of the airplane's performance are classified for security reasons, but it is reported to be one of the most advanced interceptors being produced.

A single-seater which will not carry a radar observer, the F-102 will be piloted automatically except on take-off and landing. It has been called the nearest thing to a guided missile.

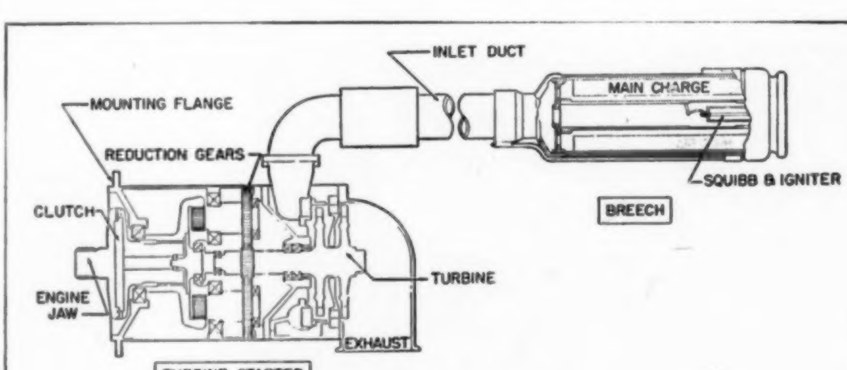
Convair built the world's first delta wing airplane, the Air Force XF-92A research interceptor, which was first flown in 1948. This aircraft will be taken over by the National Advisory Committee for Aeronautics for research studies to substantiate ultrahigh-speed airflow theories.

Air Force XF-92A research interceptor from which the new F-102 supersonic interceptor incorporating delta-wing design was developed

Small Gas Turbines Start Jets

Small, self-contained airborne gas turbines for starting jet engines eliminate the need for ground power units and are invaluable when ground power is not readily available, such as in advance combat fields. One of two types of starters which have been developed by the General Electric Aircraft Gas Turbine Div. to meet varied engine requirements is powered by hot gases resulting from the burning of a solid propellant in a repleaceable cartridge.

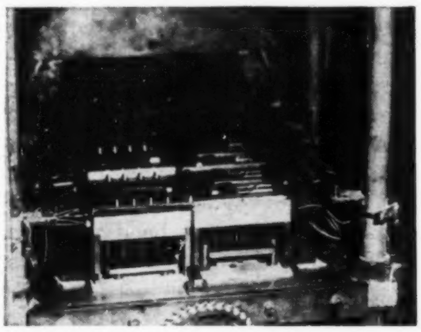
The fast-burning charge, which resembles an artillery shell, is inserted in the breech and ignited electrically. Temperatures of the gases resulting from rapid burning of the main charge range from 1800 to 2000 F; pressures range from 500 to 1500 psi. Burning time of the cartridge can be adapted to match engine requirements. When cartridge combustion is complete, the starter automatically declutches. Torque output is essentially constant throughout the operating cycle. The breech assembly can be designed to hold supplementary cartridges if desired, and each chamber can then be fired successively without reloading between shots.



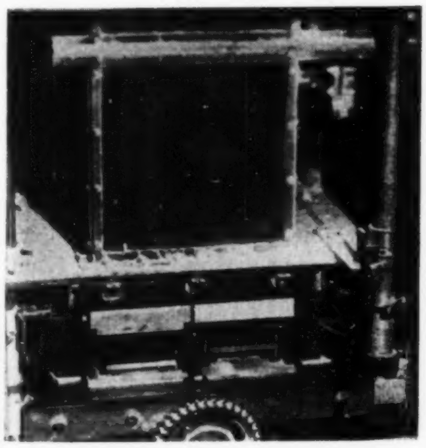
New Equipment Offered For Shell Molding

Equipment and procedures are being made available so that foundries may take advantage of the new shell molding process without heavy expenditures for experimental and development work or equipment.

Shell molding, a recently developed casting method employing a mixture of sand and thermosetting resins baked on a precision pattern to form a permanent smooth-finish mold, claims the advantages of: precision, since limits of ± 0.0025-inch per inch can be maintained; normal finish of approximately 125 microinches; ease of mold handling since cured shell molds are permanent and can be stored indefinitely without loss of accuracy.

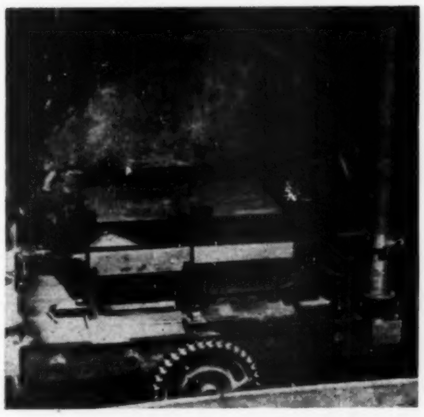


Dies in position for molding

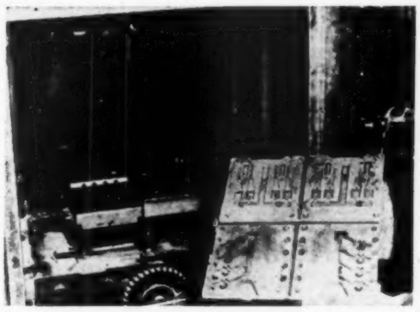


Hopper lowered over die

Precision dies and toolmaking facilities are essential to the new process. Powdered Metal Products Corp. of America, where early experimental shell molding was done, has developed a new automatic machine which can turn out approximately 20 complete shells per hour from a single set of dies. The machine performs all steps of the operation, delivering finished shells ready for pouring or storage. It will accommodate dies as large as 12 by 18 inches.



Hood in place to cure shell



Completed shell removed from die

After dies have been inserted, the sequence of operations performed by the machine is as follows:

1. Die is heated by means of electrical heating elements inserted in the die shoe.
2. The mixture of sand and resin is dumped onto the heated die from the machine hopper.
3. Contact surface of the shell is cured by holding the mix in contact with the heated die for a timed interval.
4. Excess sand mixture is dumped back into the machine hopper.
5. The shell is oven cured to make it permanent.
6. Completed shell is ejected.

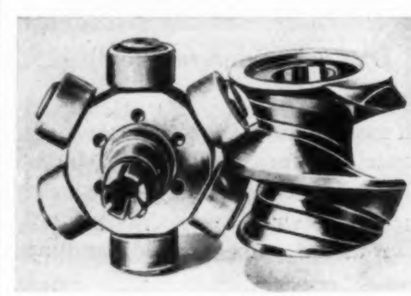
Because the molds are light and can be stored or transported easily, mold-making and pouring operations can be housed separately.

Engineering News

Market High-Speed Roller Gear Drive

Now being produced for companies which previously used Geneva drives and other forms of intermittent motion equipment, the Ferguson Machine & Tool Co. roller gear drive was designed originally for the Universal Match Corp., parent organization of Ferguson. Test results show that this drive will operate 8000 hours, or the equivalent of four shift-years, without maintenance when extreme precision is required. If extreme precision is not required, the drive will operate as many as 20,000 hours, or ten shift-years, without maintenance. The unit can be renewed after such use by replacing the standard ball bearings.

Applications of the drive include indexing dials, indexing carriers, indexing conveyors and indexing mechanisms. It has been used on dials and roll feeds, carrier chains



and conveyors at speeds as high as 800 pieces per minute to extremely close tolerances and without auxiliary locating mechanisms. It can be used for punch press dials with a precision of ±0.001-inch for 12-inch diameter dials.

Number of stops for the drive is variable from two to infinity. The indexing period can vary from 25 to 100 per cent of the total cycle time, and rotation can be either right or left-hand.

•

A new manufacturing plant to serve East Coast customers has been opened by **Parker Rust Proof Co.** at Mountain View, N. J. The company's eastern office, formerly located in New York City, is also now at Mountain View.

(*Continued on Page 237*)

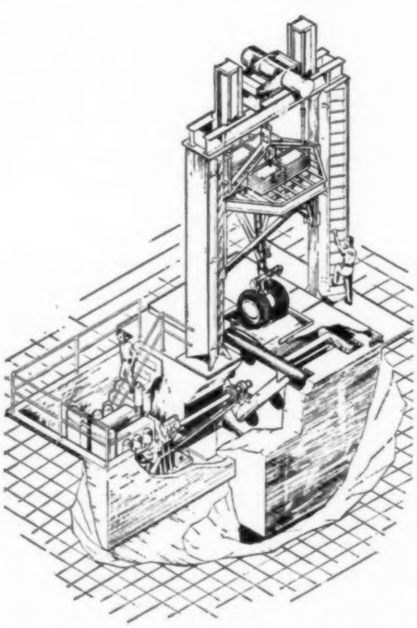
Engineering News

(*Continued from Page 234*)

Treadmill Simulates Airplane Runway

Rather than reinforce landing gears to withstand vibration, Lockheed Aircraft Corp. designers plan to eliminate it entirely. To aid in this project, a 230-ton treadmill "wobble stopper" has been built which will simulate takeoff and landing of planes as large as the Super Constellation and as fast as future jet transports.

Airplane nose landing gear units are mounted on a 21-foot tower, subjected to as much as 30 tons of weight and then run on a cylindrical drum rotating at speeds simulating takeoff, landing, or taxi operations. Periphery of the rotating drum can be provided with built-in bumps for reproducing taxi characteristics on rough airstrips.



The treadmill technique will make possible testing at higher speeds than is possible by towing weighted landing gears across runways behind high-speed automobiles, which was the method used for developing P-38 Lightning and Constellation nose gears.

Specifications of the treadmill include: diameter of drum, 10 feet; surface speed, 150 mph maximum; torque available, 3500 ft-lb (50 to 150 mph); power available, 280 horsepower at 150 mph; normal reaction force, 60,000 pounds.

Engineering News

New Conveyor Belt Is Virtually Rip-Proof

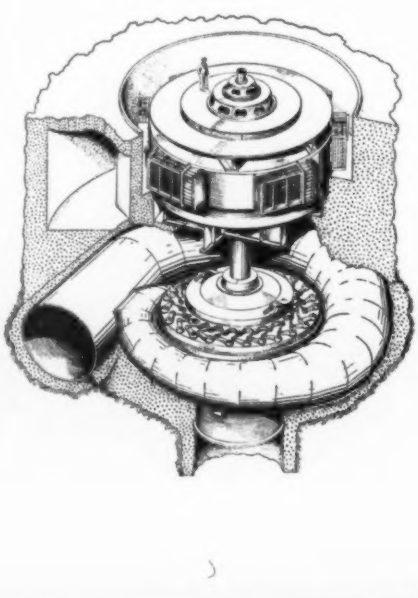
Featuring a special carcass which has multiple strands of high-tensile steel wires imbedded at three-foot intervals, a new type of conveyor belt claimed to be virtually rip-proof has been placed on the market by New York Belting & Packing Co. The belt was designed particularly for use where it might be subject to severe ripping or tearing action by foreign objects, such as in coal mining and handling, quarry work and mining of metallic ores. If the belt should be punctured, tearing is limited to approximately three feet lengthwise by the steel wire construction.

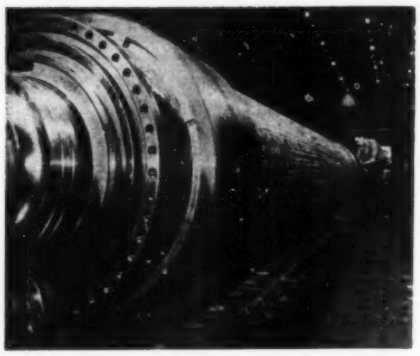
To Build Largest Motor and Reversible Pump Turbine

The largest electric motor and reversible pump-turbine ever built will provide power for a pump-storage project at Hiwassee Dam in southwestern North Carolina, part of the Tennessee Valley Authority's power network.

In this installation a single hydraulic machine will operate in one direction as a turbine and in the reverse direction as a pump. A direct-connected electrical machine will serve as a motor for pump operation and as a generator for turbine operation.

Water from the reservoir will drive the unit as a turbine-generator and add needed energy to the TVA system in peak demand periods. During off-peak periods, the





This generator rotor will be turned by a turbine which operates with the highest steam temperatures ever used in a turbine-generator. Operating at an initial temperature of 1100 F, the 145,000-kilowatt unit is one of the first 3600-rpm machines to generate electricity at 20,000 volts. The boiler for the turbine will employ coal, oil or gas, or a combination of all three fuels. Steam is used at a pressure of approximately 2350 psi. Because of the high temperature, the inner shells of the turbine are made of stainless steel. Built by General Electric for the Kearney, N. J., generating station of Public Service Electric and Gas Co., it weighs 100,000 pounds

♦

unit will operate as a motor-driven pump to refill the reservoir.

The huge pump-turbine, operating as a turbine, will have a maximum rating of 120,000 horsepower. When motor driven, the unit will have a pumping capacity of 3.3 billion gallons of water per day, nearly three times the amount required for New York City. The pump has over three times the capacity of each of those which serve the Grand Coulee irrigation project, which are at present the world's largest.

As a motor, the electrical motor-generator will be rated at 102,000 horsepower, 106 revolutions per minute. As a generator it is rated 70,000 kva, 13,800 volts. Being built by Allis Chalmers, the equipment is scheduled to be completed late in 1955.

♦

Robert S. Sweeney has been named vice president and general manager of the **Watson-Stillman Co.**, the hydraulic press division of **H. K. Porter Co. Inc.**, Roselle, N. J. Mr. Sweeney, who has been associated with Watson-Stillman since 1944, will be in charge of all manufacturing and sales activities of the division.

Engineering News

Giant Forging Presses To Be Installed

Two of the world's largest forging presses will be installed in the Cleveland works of Alcoa. They will be used primarily to make structural aluminum and magnesium forgings for high-speed aircraft.

Scheduled to be put into operation in 1954, the presses will be housed in a structure equivalent in height to a ten-story office building. The 50,000-ton capacity press will have an overall height of 80 feet, 36 feet of which will be below ground level. The 35,000-ton press is 38 feet wide, 65 feet long and will stand 49 feet above the ground. Steel reinforced concrete foundations extending 30 feet underground and resting on pilings reaching 75 to 80 feet downward will be needed to support this heavy machinery.

◆

Headquarters for the sales, engineering and production of Multi-V-Drives and Allspeed drives have been consolidated at the Oil City, Pa., plant of **Worthington Corp.,** Harrison, N. J. A new division, the Mechanical Power Transmission Div., has been set up to handle these products, and approximately 150,000 square feet of manufacturing and warehousing space has been allocated to the operation.

"I don't know why the boss hired him. He doesn't even have an engineering degree."

Huge Concrete Saucer Will Support Atomic Reactor

Foundation for a huge 225-foot steel sphere which will house a nuclear submarine power plant is 179 feet in diameter and 42 feet deep. This Atomic Energy Commission project for the U. S. Navy is under the direction of the Knolls Atomic Power Laboratory operated by the General Electric Co. The reactor itself and the sphere enclosing it are being built near Schenectady, N. Y.

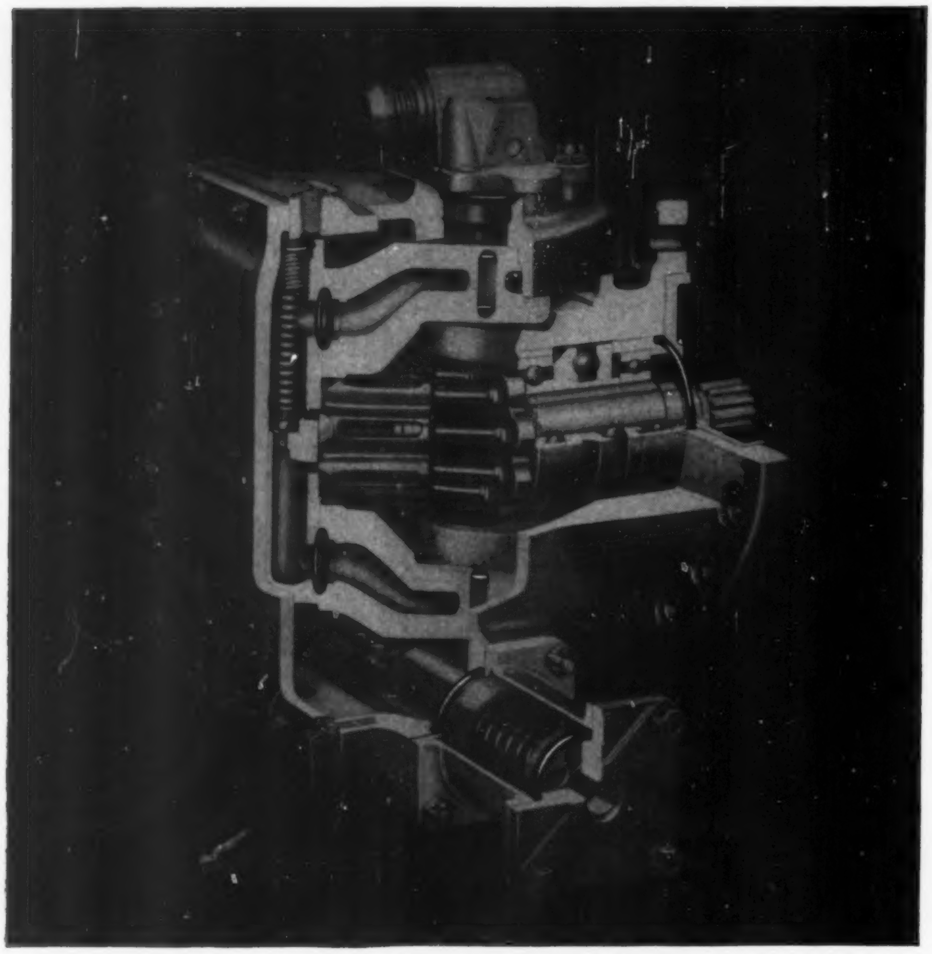
Spherical design was adopted for the reactor building to provide additional protection to operating personnel and off-site areas during test operations beyond the safety controls of the reactor itself. Welded steel plates

hoisted into position by a derrick which reaches 424 feet above ground level will form the "skin" of the structure. Each weld must be X-rayed to detect cracks.

This reactor will be used in one of two nuclear science studies being made to reach a solution to the problem of utilizing atomic fuel for underwater ship propulsion.

•

Three new sales appointments have been announced for the Machine Tool Div. of **Sundstrand Machine Tool Co.,** Rockford, Ill. George Seeburg, formerly sales engineer, has been promoted to the position of assistant general manager. T. B. Buell has been named general sales manager in charge of overall sales policies and is succeeded in his former position as sales manager by Harry Leber, former manager of direct sales.



Cutaway view showing O-rings. Illustration courtesy Vickers, Inc.

PARKER O-RINGS help pumps set endurance record

Vickers reversible flow 3,000 PSI hydraulic pumps, equipped with PARKER O-rings, recently operated 304,578 pump hours without malfunction for a 12-month period at one airline.

This noteworthy performance—in all sorts of weather—demonstrates the leakproof, long service qualities of PARKER O-rings . . . precision-molded from superior synthetic rubber compounds. Important, too, they provide simplified as well as efficient sealing. Design involves only a small groove to retain the ring. They are economical to use, easy to replace.

PARKER is the one source for *all* standard O-rings for fuel, hydraulic and engine oil services, and for special service O-rings. Ask your PARKER Distributor for Catalog 5100, or write The PARKER Appliance Company, 17325 Euclid Avenue, Cleveland 12, Ohio.

THIS IS IT

Cross section drawing of O-ring in groove, sealing under pressure.

Parker

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Los Angeles 22, Calif.

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Allrubber Products & Supply Co.
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Minneapolis 5, Minn.

NEW YORK, N. Y.
Durham Aircraft Service, Inc.
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Nielsen Hydraulic Equipment, Inc.
298 Lafayette St., New York 12, N. Y.

PHILADELPHIA, Pa.
Goodyear Supply Co.
1506 Latimer St., Philadelphia, Pa.

PORTLAND, Ore.
Hydraulic Power Equipment Co.
2316 N. W. Savier St., Portland 10, Ore.

ST. LOUIS, Mo.
Metal Goods Corp.
5239 Brown Ave., St. Louis 15, Mo.

TULSA, Okla.
Metal Goods Corp.
302 North Boston, Tulsa 3, Okla.

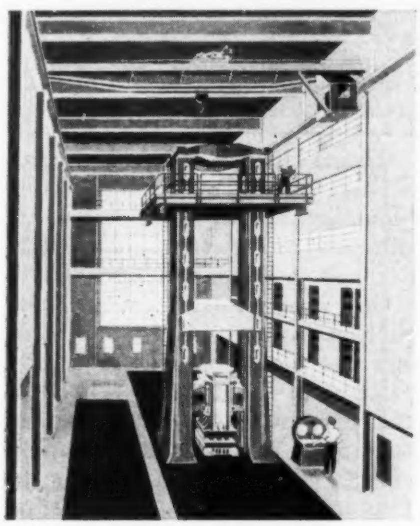
WICHITA, Kan.
Standard Products, Inc.
650 E. Gilbert, Wichita 11, Kan.

CANADA
Railway & Power Engineering Corp., Ltd.

Engineering News

Largest Vertical Universal Testing Machine Planned

Testing actual structural members rather than small replicas will be possible with the 5,000,000-pound capacity hydraulic tension-compression machine to be built at Lehigh University. Approximately 58 feet high, the machine will permit the testing of building columns and the compressed members of railroad and building trusses. Tests



will be possible on members 40 feet long.

Plans are being made to arrange for the application of auxiliary lateral loads to a specimen in the machine, and consideration is being given to the possibility of utilizing the frame of the machine in other tests.

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The new Union, N. J., plant of the Airco Equipment Mfg. Div. of **Air Reduction Co. Inc.** has been officially opened. Manufacture of welding and cutting torches, tips, regulators, oxygen and acetylene manufacturing and distribution equipment, gas-arc welding apparatus and oxyacetylene cutting machines has begun.

Report on Materials

Despite the steel strike, the iron and steel industry's output for the year amounted to 93 million tons, according to a year-end statement

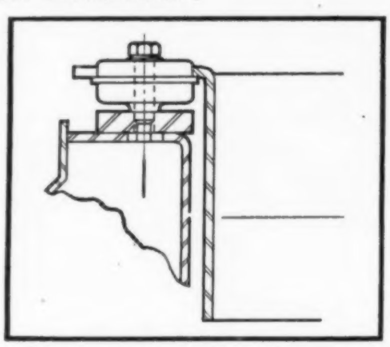
Advertisement

Lord Mountings Help Reduce "Down-Time" and Service on LeTourneau Earthmover

On "round-the-clock" operations on large construction jobs, "down-time" and service work become expensive to the contractor.

Of necessity, LeTourneau Earthmovers are operated under the most rugged conditions. It was found that rough terrain, rocks and other obstacles damaged radiators and water systems as a result of high shock loads exerted on those component parts.

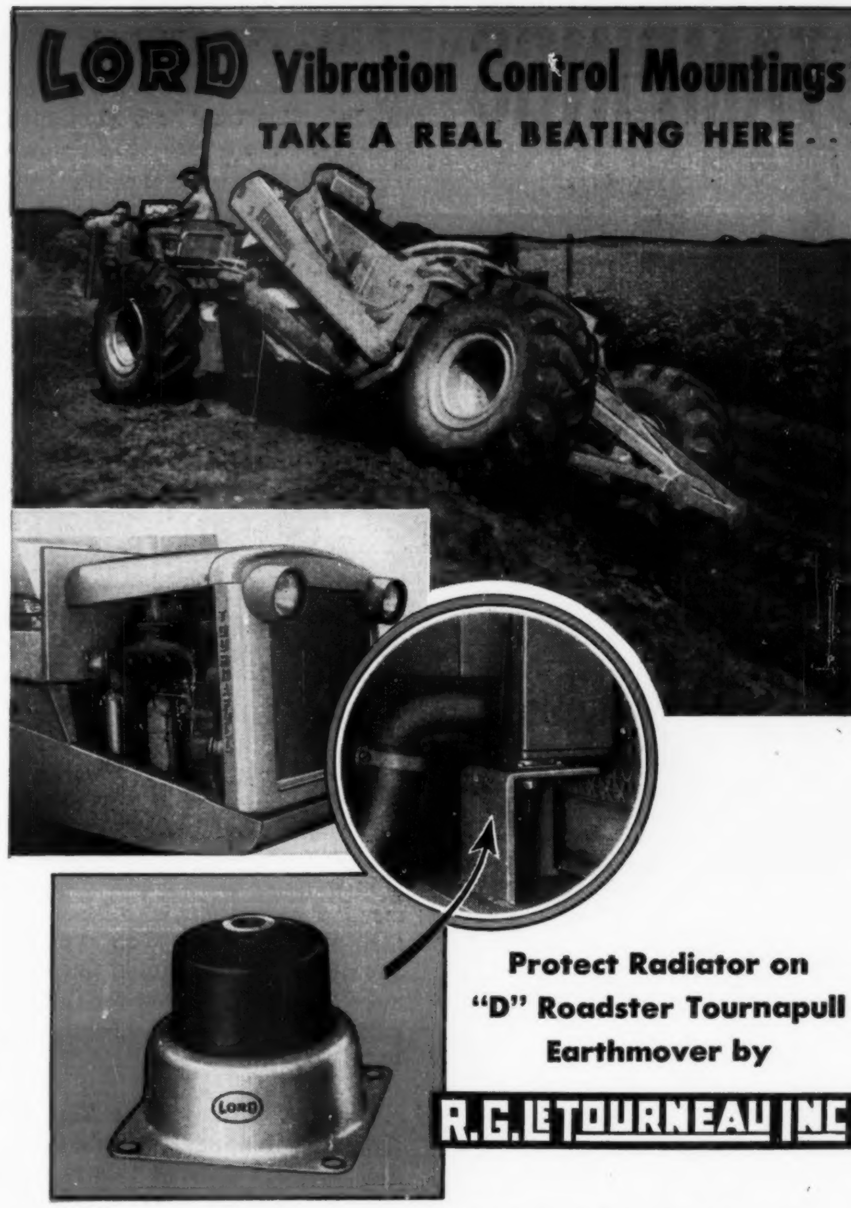
Lord plate form mountings are now used on the top of the radiator in either horizontal or vertical position, according to the construction characteristics of various LeTourneau Earthmover Models. These are shown below.



The base of the LeTourneau radiator rests on the Lord plate form mounting illustrated below.

This combination results in complete control from any direction, of engine vibration and the heavy shocks encountered in earthmoving operations. Thus, the vitally important cooling system of LeTourneau heavy duty earthmoving equipment is protected by Lord Vibration and Shock Control Mountings. Thus, also, the costly servicing of cooling systems and the resulting "down-time" have been reduced to a minimum.

Further details on the isolation of vibration and absorption of shock loads in other industrial applications are available on request from Lord Manufacturing Company, Erie, Pa.



Wherever heavy machines change the face of the earth by moving tons and tons of dirt, it's a safe bet that the "D" Roadster Tournapull Scraper is doing a giant's job. Contributing mightily to its continuous, smooth, cool running are the Lord Vibration and Shock Control Mountings which protect the radiator from the shocks this rugged "bear for punishment" must withstand day in and day out. The "D" Roadster's Lord Mountings absorb the twisting, tearing strains of heavy duty earthmoving. The cooling system vital to its uninterrupted operation functions perfectly no matter how rough the going.

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by the American Iron and Steel Institute. This total was 12 million tons below the record 1951 production, but production capacity at the end of the year was about 116 million tons.

Direct defense requirements for steel in 1953, including atomic energy requirements, are not expected to exceed 14 million tons, or about 12 per cent of the anticipated output of 119 million tons. The Steel Products Industry committee of NPA recently recommended immediate open-ending of the Controlled Materials Plan as it affects steel, in accordance with the report of a task group which studied the demand-supply situation in the steel products industry. Members declared that, after taking care of military demands, there would be no significant dislocation of steel distribution to the civilian economy, and steps should be taken immediately to begin orderly revocation of Government controls.

In essence, the committee recommended revocation of all nonmilitary CMP authority on April 1, and that no applications for allotment authority be filed for the second and subsequent quarters of 1953.

Second-Quarter Steel: The Defense Production Administration, however, has ignored this request and has allotted steel for the second quarter, increasing the amount to makers of automobiles and other civilian products to 70 per cent of the amount consumed before the outbreak of the Korean war. This is the highest allotment of civilian steel so far, being about 16 per cent higher than the third quarter of 1952. Copper and aluminum, by comparison, will be held at about present levels.

Considering this situation, it seems highly unlikely that controls will be affected before the cutoff date of June 30 (MACHINE DESIGN, December, 1952, Page 262).

Boron Steel Output Rises: Production of lean-alloy boron steel has risen rapidly in 1952. In the first nine months, output was 507,000 tons, far exceeding the complete output in 1951 of 354,000

tons. Expressed in percentage of total alloy steel other than stainless, production jumped to 9.2 per cent from 4 per cent in 1951. Conservation of critical alloying elements and difficulty in obtaining certain types of alloy steels seems to have accounted for the quick growth, since the use of only 8 pounds of boron in a 150-ton heat of steel saves a total of 4800 pounds of critical nickel, chromium and molybdenum.

Drought Holds Aluminum Down: Power shortages cut production of primary aluminum for the second straight month in October, according to the latest available figures of the Aluminum Association. The continuing drought in the Pacific Northwest held total production down to 155 million pounds, compared with 154 million pounds for September and 170 million pounds for August. At the same time, capacity by the end of 1953 is expected to be about 250 million pounds a month, which may offset some present effects of the shortage.

Plastics Continue at High Level: If the present trend in the plastics industry holds, a record level of activity will be established during the first quarter of 1953. Starting from a low level during the summer months of last year, plastics molders are currently very busy. Some, however, feel that there are signs of growing surplus, both in materials and in available production time.

Cold Rubber Preferred: According to the Reconstruction Finance Corp., there has been an increased preference for synthetic rubber produced at low temperatures, primarily because of its increased durability for tires. Estimates received by RFC for the last quarter of 1952 and the current quarter indicate that manufacturers of synthetic rubber (GR-S) products will purchase 174,000 tons of cold rubber and 316,000 pounds of hot rubber. Additionally, about 34,000 tons of butyl (GR-I) rubber, principally for the manufacture of inner tubes, will be used.

Meetings

AND EXPOSITIONS

Jan. 12-15—

American Management Association. General management conference to be held at Hotel Statler, Los Angeles, Calif. Additional information may be obtained from society headquarters, 330 West 42nd St., New York, N. Y.

Jan. 12-16—

Society of Automotive Engineers. Annual meeting and engineering display to be held at the Sheraton-Cadillac Hotel in Detroit, Mich. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Jan. 19-22—

Plant Maintenance Conference. Fourth annual conference to be held at the Public Auditorium in Cleveland, Ohio. Additional information may be obtained from the exposition management, Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Jan. 19-23—

American Institute of Electrical Engineers. Winter general meeting to be held at Hotel Statler, New York, N. Y. Additional information may be obtained from society headquarters, 33 West 39th St., New York 18, N. Y.

Jan. 21-23—

Society of Plastics Engineers. Ninth annual technical conference to be held at the Hotel Statler in Boston, Mass. Additional information may be obtained from society headquarters, 409 Security Bank Bldg., Athens, Ohio.

Jan. 23—

Malleable Founders' Society. General society meeting to be held at Hotel Cleveland, Cleveland, O. Additional information may be obtained from society headquarters, 1800 Union Commerce Bldg., Cleveland, O.

Jan. 26-30—

International Heating and Ventilating Exposition. Eleventh ex-

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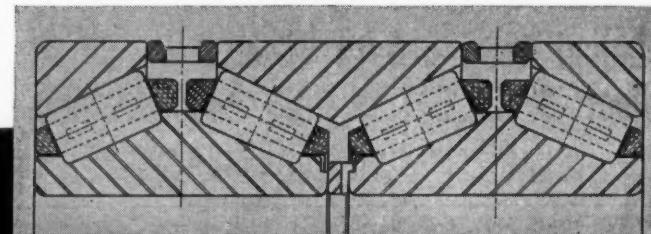
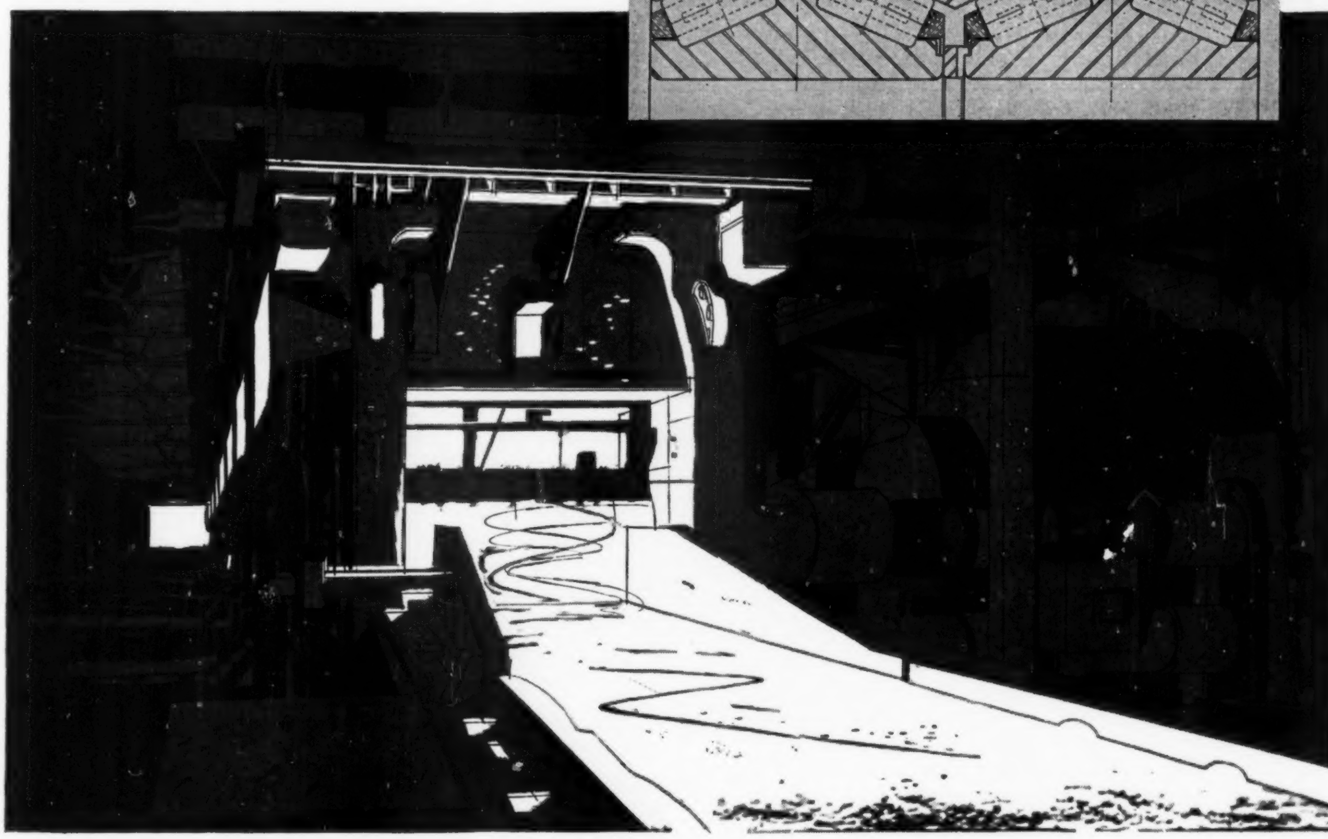
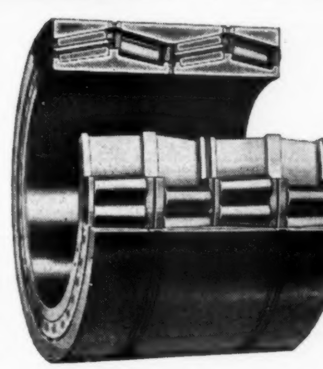
TORRINGTON Four-Row Tapered Roller Bearings on finishing-stand work-rolls of U. S. Steel 80" hot-strip mill were specially designed for high thrust loads.





Rolling to a new world's record!



2,621,500 net tons of 2000-foot coils of uniform strip! That was the world's record rolled in 1951 by U. S. Steel's 80" hot-strip mill—on TORRINGTON Tapered Roller Bearings.

Precision-ground TORRINGTON Tapered Roller Bearings are used on many of the work-roll and back-up thrust positions, from the huge mill's No. 1 scale breaker to its finishing stands.

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Meetings and Expositions

position to be held at the International Amphitheatre in Chicago, Ill. Charles F. Roth, 480 Lexington Ave., New York 17, N. Y., is manager.

Feb. 4-6—

Western Computer Conference. First meeting to be held at Hotel Statler, Los Angeles, Calif. will be sponsored by the joint Computer Conference Committee of the Institute of Radio Engineers and the American Institute of Electrical Engineers. Additional information may be obtained from Co-chairman Gilbert McCann, California Institute of Technology, Pasadena, Calif.

Feb. 16-18—

American Management Association. Personnel conference to be held at the Palmer House, Chicago, Ill. Additional information may be obtained from society headquarters, 330 West 42nd St., New York, N. Y.

February 18-20—

Society of Plastics Industry. Eighth annual reinforced plastics division conference to be held at the Shoreham Hotel, Washington, D. C. Additional information may be obtained from society headquarters, 67 West 44th St., New York 18, N. Y.

Mar. 2-6—

American Society for Testing Materials. Spring meeting to be held at the Statler Hotel, Detroit, Mich. Additional information may be obtained from society headquarters, 1916 Race St., Philadelphia 3, Pa.

Mar. 3-5—

Society of Automotive Engineers. Passenger car, body and materials meeting to be held at the Sheraton-Cadillac Hotel, Detroit, Mich. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Mar. 17-18—

Steel Founders' Society. Annual meeting to be held at the Edgewater Beach Hotel, Chicago, Ill. Additional information may be obtained from society headquarters, 920 Midland Bldg., Cleveland, O.

Design Abstracts

(*Continued from Page* 176)

in nonsoap gelling agents for greases, and this trend has been accelerated by a threatened shortage in lithium soaps due to an increased demand by the Armed Services for lithium-soap greases. New gelling agents for greases that have been particularly interesting are colloidal silica, coated bentonite, copper phthalocyanine, various special forms of carbon black, and coated silica; and the most recent arrival in this field is treated Attapulgite.

The advent of these new gelling agents has brought new problems. Both silica and bentonite when uncoated were too sensitive to the presence of water to be useful in many applications of greases. That difficulty was cured by coating the particle with a water-repellent film; however, the resulting greases were deficient in rust preventive properties. When polar-type rust inhibitors were added to these greases, there resulted a decrease in the effectiveness of the gelling agent and in the heat stability. Also, the water-repellent films on the particles of the gelling agents appear to oxidize gradually at 250 F or higher, and the greases gradually become hard and caked. Such difficulties still limit the applications of these greases at high temperatures.

Represented in bar chart form in *Fig.* 1 are the temperature ranges of operability for the more useful of the greases developed in the past 10 years from synthetic oils and the newer gelling agents. Some of the gains obtained by diesters or diester-petroleum blends and lithium or other soaps are shown in the lower half of *Fig.* 1. Petroleum-base products, which were useful around 250 F and also at −20 to −40 F, were of very limited value at −65 F. Others useful at −65 F were of little value above 175 F. In contrast, greases made from diesters, or from diesters containing a minor proportion of petroleum, have excellent, long-life, performance characteristics over the temperature range from 250 to −75 F in some in-

stances and from 250 to −100 F in others. Furthermore, the former group of ester-base greases have a more limited, but useful, service life at 300 F as indicated by the darkened area at the right end of the bar chart.

In the upper half of *Fig.* 1 are given the temperature ranges for the new, synthetic, high-temperature greases. The large gains at both extremes of temperature resulting from the development of the lithium soap-silicone greases is evident in the horizontal bar from the top. Earlier AN-G-5A petroleum greases were useful from −50 to 300 F with a more limited life up to 400 F. The phthalocyanine gelling agents have greatly widened with the temperature range of greases in general, whether they are made from silicones, diesters, fluorocarbons or petroleum. *Fig.* 1 also shows the gains made possible by gelling silicones with copper phthalocyanine. When the silicone liquid used is a methylphenyl silicone, the temperature range possible is from −20 to 450 F, and a more limited life is obtainable at 500 F. Long operation in ball bearings at temperatures much above 250 F requires the use of specially made bearings. When the methyl silicone liquids are used, the resulting phthalocyanine greases are useful from −100 to 400 F, and they have more limited usefulness at 450 F.

Future Holds Promise

Continued progress can be predicted in the improvement of properties of greases such as their work and storage stability, their resistance to oxidation, and their temperature range of operation. Wider temperature ranges, and much longer storage and operating lives in grease lubricated systems can be expected with the adoption of properly manufactured greases made from synthetic oils. Indicated improvements in greases, antifriction bearings, and in the end-bell design of electrical motors will make it possible to approach much closer to the engineers' goal of lifetime lubrication for equipment operating in the temperature range of −65

Design Abstracts

to 250 F. At operating temperatures of 300 F the relubrication intervals will eventually be lengthened to 5,000 hours, while at 450 F it will be at least 1,000 hours. The increased cost of such new greases, and the necessary bearings to be used with them, should be more than justified by the resulting improved performance and the decreased cost of maintenance and repair.

Recent Applications: With the advent of gas-turbine-powered aircraft during World War II, better lubricants were soon needed to make the most of the performance capabilities of these new engines. Since the military value of turbojet engines was in their rapid rate of climb and their high speed, a lubricant was needed which would be sufficiently fluid to permit starting at subzero temperatures without dilution or the use of heaters. It was also necessary to retain enough viscosity at engine operating temperatures to support large bearing loads. In combat, extremes of temperature frequently range from −65 F (or lower) to main bearing temperatures in flight of from 285 F to as high as 400 to 500 F soon after engine shutdown. The upper temperature is caused by heat flow to the bearing, which had been largely dissipated in flight. Hence, the oil had to be unusually stable to heat and oxidation. Low volatility at elevated temperatures was essential, since evaporation of the oil was increasing oil consumption and was leaving the bearings unlubricated after shutdown. Furthermore, evaporation of the less viscous and more volatile fractions in some oils increased the viscosity of the residual oil and increased the power required for low temperature starts.

The first lubricants used in gas-turbine-driven aircraft were petroleum oils. Much lower viscosity grades were adopted a few years after World War II; resulting viscosity grades 1010 and 1015 were listed in 1947 in specification AN-O-9 and again in 1950 in MIL-O-6081. Since grade 1015 oil had a pour point of only −50 F and a viscosity of 20,000 centistokes at −45

F, it was unsuitable for most military aircraft. It was evident soon after World War II ended that such lubricants as the diesters, and possibly the doubly chain-stoppered polypropylene oxides, when used either alone or in mixtures could extend greatly the temperature and altitude range of turbojet-powered aircraft. Not only had these diester oils larger viscosity indices but also much smaller viscosities and volatilities at −65 F than grade 1005 oils. In fact, some diester oils have the viscosity at 210 F of grade 1010 oil, at −65 F of grade 1005 oil, and yet they are able to satisfy the specification requirements of grade 1010 oil relating to flammability and volatility.

Additives Widen Service

One requirement of the Armed Services was that a single specification should care for a lubricant for turbojet engines as well as for the reduction gear systems of turboprop engines. Gear tests proved it was necessary to incorporate an additive in the diester oil in order to support the high unit loads developed. Many of the conventional types of additives could not be used because the same gear oil had to lubricate the antifriction bearings of the turbine and the supporting bearings of the gears. Tricresyl phosphate was the first acceptable wear preventive found which would increase the load carrying capacity of diester oils without corroding bearings or causing deposits. However, its adoption considerably raised the viscosity at −65 F.

This led to the adoption of specification MIL-L-7808 for a synthetic lubricating oil for gas-turbine-powered aircraft engines. It is interesting to compare the requirements of grade 1010 petroleum oils with those of MIL-L-7808. The ester oil is slightly more viscous at 210 F and also at 100 F; yet it is only one-third as viscous at −65 F. The flash point of the ester oil is 150 F higher than that of 1010, and yet it is only 1/80 as volatile. In preparing this specification, there was some doubt about the completeness of present knowledge of how to specify such radically different air-

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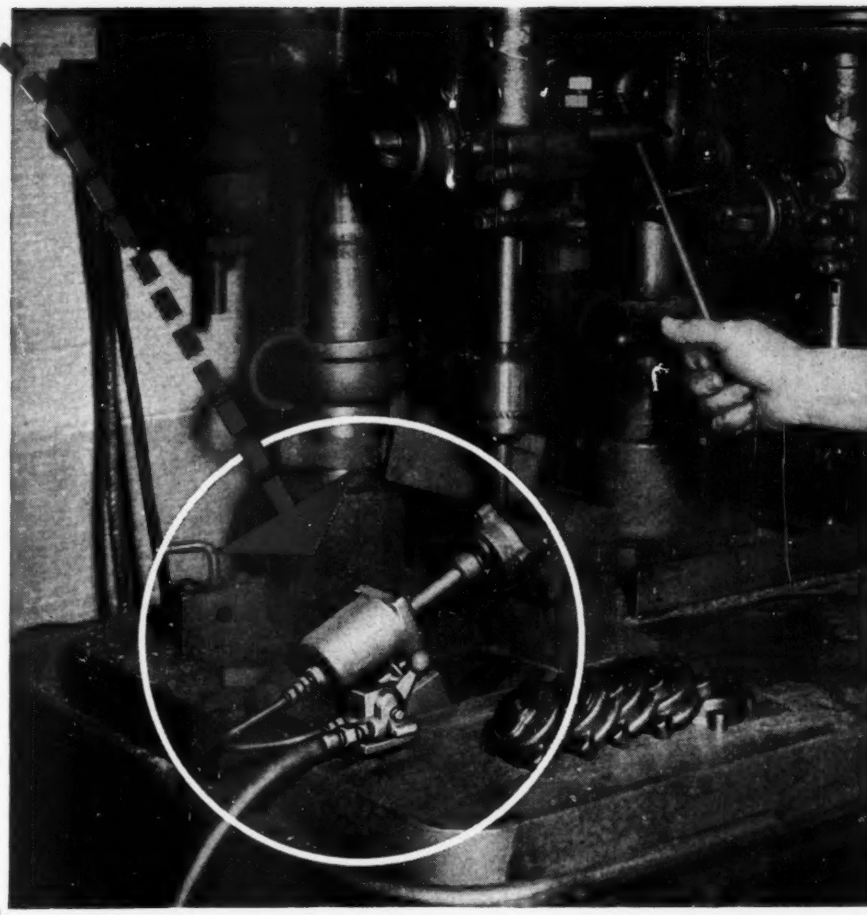

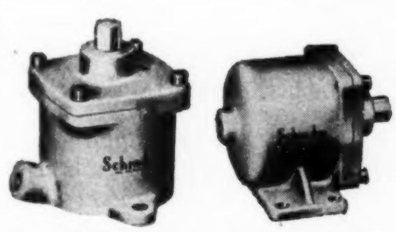

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craft engine oils. Hence, the devise was used of citing in the new specification the chemical composition of a reference oil which would be used to establish any changes in the specifications that service experience should make necessary. Unfortunately, many people have misunderstood this point and have assumed that the Armed Services were departing from their traditional use of performance specifications for materials. As more efficient antiwear agents are found, and as the availability of the newer esters increases, it may become possible to lower the maximum permissible viscosity at −65 F considerably below the present specified value of 3,000 centistokes. The requirements of MIL-L-7808 are by no means as severe as laboratory data indicate they could be, and many synthetic esters exist which are capable of surpassing the specified viscometric characteristics. However, the present specification was based on the immediate availability of a few of the aliphatic diesters.

Maintenance Reduced

Another important application of synthetic oils and greases is in the lubrication of aviation instruments. During World War II there was much difficulty with the gyros used for aviation and fire control, and in some extreme instances it was necessary to overhaul them after 50 hours of flight. In general, the introduction of the synthetic oils or greases has greatly decreased the lubrication difficulties encountered in aviation and ordnance gyros.

One of the most interesting of the postwar applications of synthetic tailor-made lubricants has been in the lubrication of automatic weapons such as aircraft cannons and machine guns. The synthetic ester oils have been found so effective for lubricating these essential mechanisms over the temperature range −70 to 150 F that they are now specified by the Navy (MIL-L-17353 BuOrd) and are supplied by several producers. The diester greases have also been found ideal for lubricating the feeder mechanisms. Some of the esters

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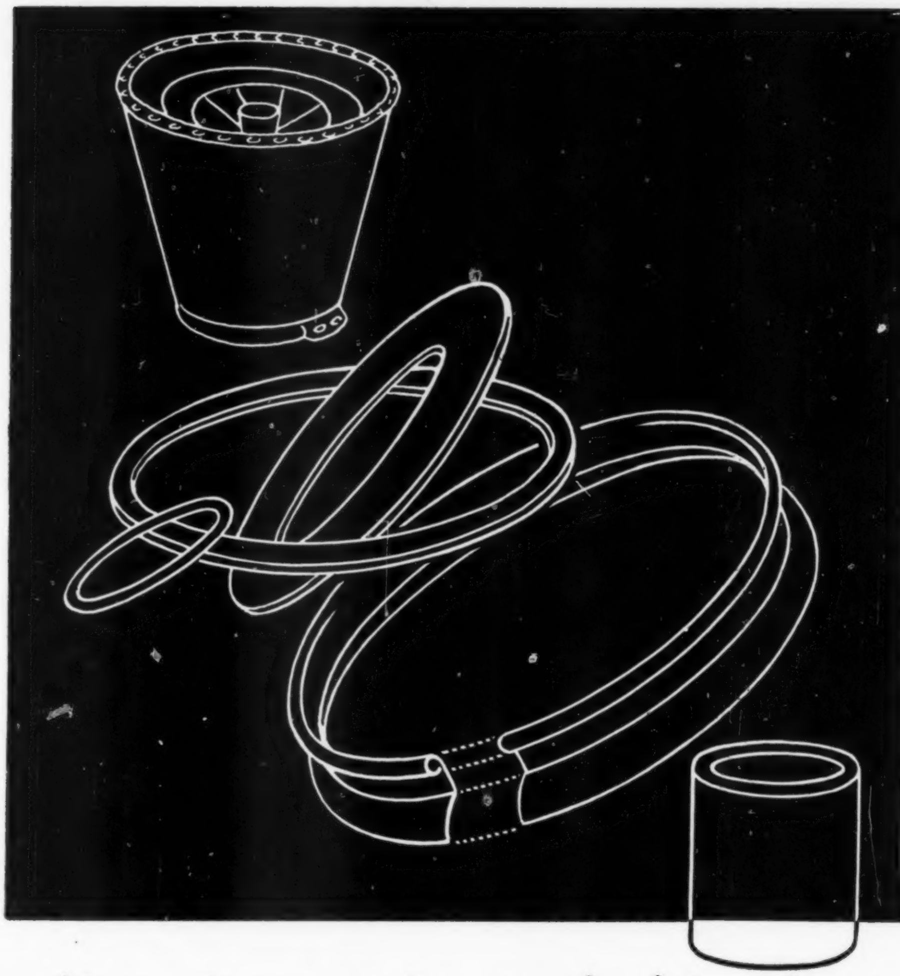

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listed in TABLE 3 have been used widely in the past year, others are of promise, and many more equivalent materials have been synthesized. Although these light-grade oils were developed specifically for various military applications, they should be valuable in lubricating a much wider variety of mechanisms which must be operated at subzero temperatures or with very limited power at ordinary temperatures.

Undoubtedly, the use of the new synthetic lubricants can be expected to broaden and become increasingly valuable to the engineer. Probably some of these materials will one day become commonplace in the shop and home for the lubrication of familiar mechanisms varying from electric motors, sewing machines, fans and typewriters to locks, sportsman's rifles, and fishing reels.

From a paper entitled "Engineering Possibilities of Synthetic Lubricants" presented at the SAE National Fuels and Lubricants Meeting in Tulsa, Okla., November 1952.

Electric Analogies for Beam Analysis

By Stanley U. Benscoter and Richard H. MacNeal
California Institute of Technology
Pasadena, Calif.

IN RECENT years, progress has been made in the development of applications of electrical-circuit analogies to problems in structural analysis. Many solutions of such problems have been obtained on an analog computer. A number of papers have also been written in which the electrical analogies for various problems have been presented. These papers have been written mainly by electrical engineers and have assumed the reader to be reasonably familiar with circuit theory. Consequently structural engineers have found some difficulty in appreciating and evaluating this new method of analysis. In this report, the design of the electrical circuits is explained without assuming any knowledge

of circuit theory on the part of the reader.

From a mathematical viewpoint the theory of circuit analogies has not been fully developed. Questions of existence and uniqueness remain yet to be answered. From an engineering standpoint the procedure for designing a circuit which is analogous to a given structure is at present dependent to some extent upon the ingenuity of the designer. It would be desirable to have available a set of rules defining a standardized design procedure which could always be depended upon to give a correct analogous circuit for any given differential equation. However, no such rules have been developed.

Circuit Equations

To develop an electrical analogy for the action of a beam under load it is necessary to replace the differential equations which govern the stresses and deflections by difference equations. The relations between voltages and current in an electrical circuit may also be expressed by difference equations. An analogous electrical circuit for a beam is one in which the voltages and currents are related by difference equations which have a form identical to that of the structural equations, including the boundary conditions. When such a circuit is designed and constructed, the physical circuit, with its components, is called an analog computer. It might also be called an electrical simulator or even an electrical model.

Several principles have been found to be useful in a large number of circuit designs and these will be stated briefly. Exact limitations upon the usefulness of the principles are not known.

The first principle is that the equations governing the action of the structure must be expressed as difference equations with respect to the space co-ordinates. If the stresses or deflections are governed by differential equations these equations must be replaced by difference equations.

The two quantities which are

available in an electrical circuit are voltages and currents. The two types of quantities which occur in structural analysis are force quantities and displacement quantities. Two general methods can be used in designing analogous circuits. In one method the voltage drops across various elements of the circuit correspond to forces and the currents correspond to displacement quantities. In the second method the voltages at nodal points correspond to displacement quantities and the currents correspond to force quantities. The second method has been found to be more efficient on an analog computer for beam and plate analogies and has been employed throughout this report.

Energies Must Balance

Another simple principle can be stated for circuits with passive elements—inductors, resistors, capacitors, and transformers. In many cases it is possible to determine the correspondence between currents and force quantities after the correspondence between voltages and displacements has been assumed. Whenever the voltages at either end of a circuit element both represent the same type of displacement quantity, the product of current and the drop in voltage across the element must correspond to a true structural energy quantity (or derivative or integral with respect to time of a structural energy quantity). The structural energy quantity may be either internal strain energy or external work.

In the equations which govern the structure there are three types of relations involved. Forces are related to forces from equilibrium conditions. Forces are related to displacements from an application of Hooke's law. Displacements are related to displacements from geometric conditions of continuity. The equations of the electrical circuits involve three corresponding types of relations. Currents are related to currents from Kirchhoff's nodal law which states that the outflow of current at a node

Above—A few of the wide variety of TEFLON packings and gaskets manufactured in the Garlock factories.

must equal the inflow. Currents are related to voltages from Ohm's law for a resistor and similar laws for an inductor or a capacitor. Voltages can be related to voltages by means of the transformer law. Voltage relations are also given by Kirchhoff's mesh law.

A final principle may be stated in relation to the coefficients of the equations which govern the electrical circuit. For maximum simplicity in the process of solution it is necessary to design the circuit in such a way that the equations governing the circuit have coefficients which are independent of the frequency.

From NACA Technical Note 2785 (*September* 1952), *"Introduction to Electrical Circuit Analogies for Beam Analysis." This report contains* 48 *pages with elementary diagrams and equations to illustrate the procedures for designing analogous circuits.*

Designing with Nylon

By Robert Lurie

Application Engineer
Chemical Div.
General Electric Co.
Pittsfield, Mass.

ALTHOUGH nylon is generally associated with such decorative items as stockings and sweaters, its versitility extends well into the fields of bearings, gears, and other workaday applications. Nylon is tough at low temperatures, is resistant to abrasion, has strength in thin sections, is light in weight, has certain self-lubricating qualities, and is resistant to many chemicals. Properties are given in TABLE 1. Exact characteristics and specifications can be found in the literature of manufacturers of nylon molding powders.

Bearing Tests: Results of tests using nylon against nylon, steel, and brass under various loads and speeds show the lowest rates of wear are found when nylon is run against nylon in a dry condition, as well as when nylon is run against steel lubricated with SAE 10 oil. When nylon is rubbed against brass, the metal flakes off and con-

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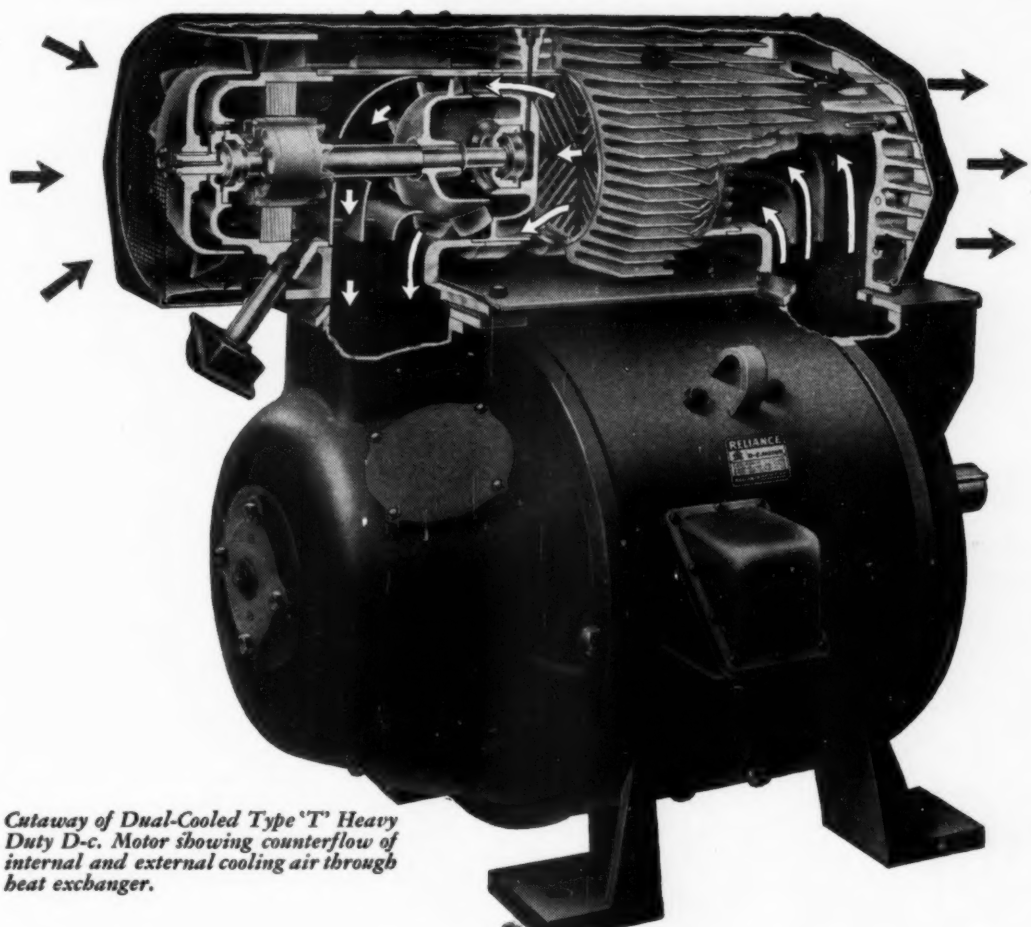

Cutaway of Dual-Cooled Type 'T' Heavy Duty D-c. Motor showing counterflow of internal and external cooling air through heat exchanger.

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taminates the bearing surface.

Rate of wear is greatest at the start but tapers off with further operation. The reason is that after the surface of the nylon is glazed more intimate contact is established, with a resultant decrease in unit pressure. This rate of wear appears to reach a minimum after about 0.001 to 0.002-inch of surface is removed. In contact with steel or brass, this figure represents the amount of nylon that will be removed in smoothing the metal to prevent further wear. The coefficient of friction decreases during the first few hours of operation and then remains constant; at the same time the rate of wear becomes practically negligible.

Table 1—Properties of Molded Nylon (FM-10001)

Property	Value
Tensile strength, 70 F	15,700 psi
77 F	10,900 psi
170 F	7,600 psi
Elongation, 70 F	1.6%
77 F	50%
170 F	320%
Modulus of elasticity 77 F	400,000 psi
Flexural strength, 77 F	14,600 psi
Creep in flexure*	90
Hardness, Rockwell	R118
Flow temperature	>480 F
Thermal conductivity	1.7 Btu/hr/sq/ft/F/in
Heat distortion temp, 264 psi	150 F
Volume resistivity	4.5 x 10^{13} ohm-cm
Dielectric constant, 60 cycles	4.1
Power factor, 60 cycles	0.014
Water absorption	1.5%
Specific gravity	1.14

*Mils deflection in 24 hours of a ½ x ½-inch bar 4-inch span, center-loaded flatwise to 1000 psi, minus the initial deflection.

When nylon is rubbing against steel, the limiting load is 550 psi without lubrication. Water lubrication will increase this limiting load 91 per cent; SAE 10 oil will increase it 182 per cent.

Wall thickness greatly influences the life of a bearing. For instance, tests showed that a bearing having a wall thickness of 0.125 inch failed after 240 hours of operation. It melted. This bearing was operated at a rubbing speed of 525 fpm, under a 220-psi load, and lubricated with SAE 10 oil. Another bearing, tested under the same conditions, but having a wall thickness of only 0.020 inch showed no sign of failure after 360 hours of operation. Re-

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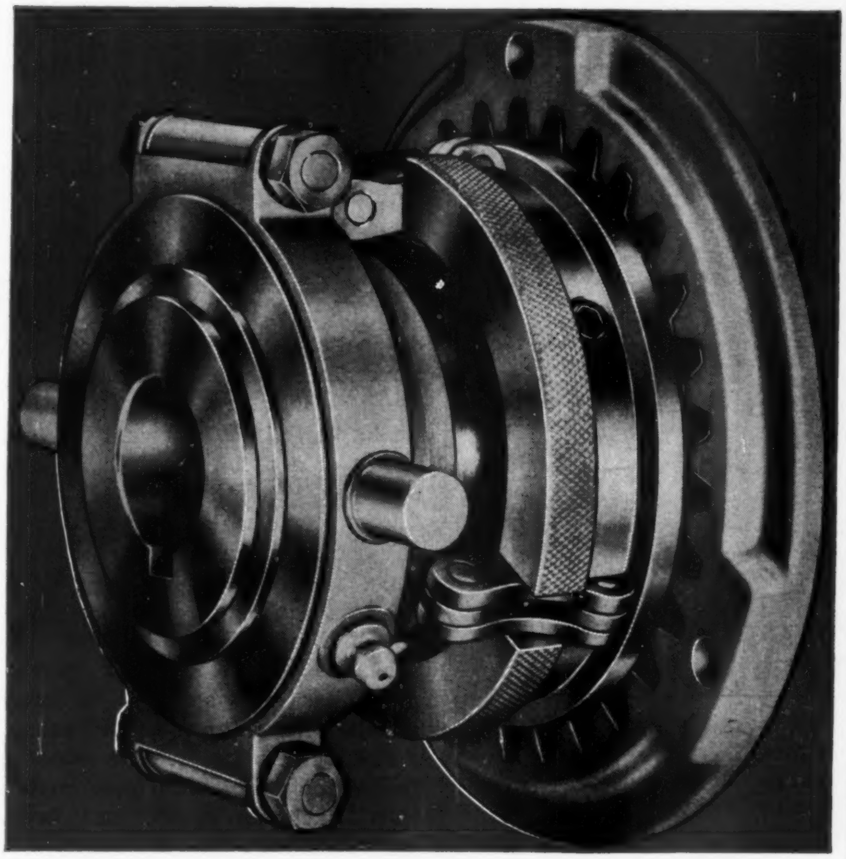
an addition to the* MAXITORQ *line



The Johnson 350 and 450 Single Disc Clutches are the newest additions to the Carlyle Johnson line... fitting companions to the Maxitorq multi-disc series.

They are ideal for light machinery service to 6 H.P. Several driving combinations are available, including V-belt. Far greater capacity at low cost is provided. (See column at right for typical applications.) They have the same "floating disc" principle as the Maxitorq Clutch ... discs that ride free in neutral ... no drag, no abrasion, no heating. A simple hex-key frees the knurled ring for easy manual clutch adjustment. Machine designers will find the solution to many problems with this new Johnson Clutch.

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ICJ53

Frankly

SPEAKING

Because of a large backlog of orders, The Carlyle Johnson Machine Company found it necessary to postpone the announcement of the new Johnson Single Disc Clutches.

Now, with added equipment for high speed production, it is possible to produce and ship the new clutches without undue delay. In fact, small orders or units for tryout in new machines will be forwarded at once. Design features make the Johnson especially suitable for installation in the following machinery:

Accessory Drives, Air Compressors, Bag Making, Boat Drives, Bread Wrapping, Chain Drives, Combines (farm), Conveyors, Crop Seeders, Cultivators, Dusting Machines, Feed Grinders, Floor Scrubbing, Fruit Cleaners, Gasoline Engines, Generator Drives, Hay Balers, Hoists, Loaders, Lawn Mowers, Milk Coolers, Mixing Machines, Motor Scooters, Packaging Machines, Paper Shredders, Power Fans, Power Saws, Power Take-Offs, Pumping Equipment, Sand Spreaders, Sewing Machines, Sheep Shearers, Spraying Equipment, Textile Machinery, Threshers (small), Tobacco Machinery, Tool Grinders, Tractors (garden), "V" Belt Drives, Vegetable Sorters.

Naturally, these are but a few of the possible applications for the Johnson Single Disc Clutch. The field is wide open, with new machinery constantly being developed.

Included in the driving combinations are: Gear Tooth, Bolted Plate, Pulley Type, Hub Adapter, Cut-Off Coupling Adapter, Single V-Belt Pulley Drive, and Double V-Belt Pulley Drive.

Carlyle Johnson engineers offer their engineering assistance in cooperation with your engineers and machine designers to develop the correct solution of your power transmission requirements. Write to Frank R. Simon, The Carlyle Johnson Machine Co., Manchester, Conn.

sults were the same when water was used as a lubricant or when no lubricant at all was used. The advantage of a thin-walled bearing is that the heat of friction is more rapidly dissipated than with a heavier bearing where the nylon acts as an insulator.

Gear Design: Where speeds or loads may be excessive, consideration must be given to the low heat conductivity of nylon, namely, 1.7 Btu per hour per square foot per degree per inch. Allowance must also be made for heat dissipation; otherwise the nylon will soften and deform because of a heat build-up. Developmental gears have been made with nylon around a copper insert to facilitate conduction of heat away from the gear-tooth area. Thus, in the more critical gear designs the problem of heat dissipation is minimized, and at the same time the beam strength of the gear is increased.

The horsepower transmitting capacity of a nylon gear can be determined through the use of the Lewis formula, usually found in any standard handbook for mechanical engineers. In the Lewis formula the safe working stress is the product of the static stress factor and velocity factor. In gear work, nylon has been assumed to have a conservative figure of 6000 psi as a static stress factor. The velocity factor for nonmetallic gear materials, such as nylon, has been worked out by the Massachusetts Institute of Technology, and is

$$K_v = \frac{150}{200 + v_p} + 0.25$$

where K_v is the velocity factor and v_p is the pitch line velocity.

The Lewis formula was originally derived for rigid metal gears where slight inaccuracies of tooth form and spacing create high dynamic loads that must be sustained by a single tooth. Under load conditions, however, the resiliency of nylon may allow contact of two or more teeth. Consequently, results obtained from this formula are conservative.

Other Applications: Nylon mold-

This is a locomotive gas turbine mid-frame. As a fabrication it consisted of a number of steel castings and rolled plate components.

Conversion to the rather intricate steel casting called for considerable ingenuity; also close cooperation between the engineering department and steel foundry.

In addition to reducing the cost 50%, redesign to a steel casting (1) resulted in a far superior product (2) cut production time two-thirds (3) substantially reduced the weight of completed structure (fabricated mid-frame shown is not completed to the same point as the steel casting) and (4) improved appearance.

* * *

Here is another example of the engineering teamwork in design and redesign of parts which is resulting in lower costs and improved serviceability with steel castings.

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Design and Build With Steel Castings

ings offer substantial savings. As a specific example, a pound of nylon, although 2.5 times as expensive as a pound of bronze, is only 0.133 as dense. Resultant saving on material is therefore, 66.8 per cent. Also further economies can be realized when the cost of injection molding versus machining operations is considered.

The translucence of thin sections of nylon, plus its ability to withstand shock, make it especially applicable to problems of illumination. A noteworth property of nylon is its ability to damp mechanical vibrations with a consequent reduction of noise. This same property is being utilized in sound-recording and motion-picture equipment.

The "conformability" of nylon—that is, its ability to deform slightly to absorb shock—makes it the choice for gears and other applications where impact loading is encountered. This property of conformability enables nylon gears to satisfactorily mesh with metal gears and to "iron out" any irregularity that may be present in gear teeth.

Nylon has been successfully applied as coil forms, pulleys, screws, valve seats, washers, pump impellers, and hydraulic needles. It continues to find use wherever toughness, abrasion resistance, strength in thin sections, and chemical resistance are required.

From an article entitled "Nylon for Bearings and Gears" appearing in the July 1952 *issue of the General Electric Review.*

Statistical Nature of Fatigue

By E. Epremian and R. F. Mehl
Carnegie Institute of Technology
Pittsburgh, Pa.

OF THE many ways in which metals can fail under the application of loads, fatigue is probably the most widespread and important. Any mechanical part which undergoes vibration, rotation, or reciprocation can suffer fatigue failure, and it has been re-

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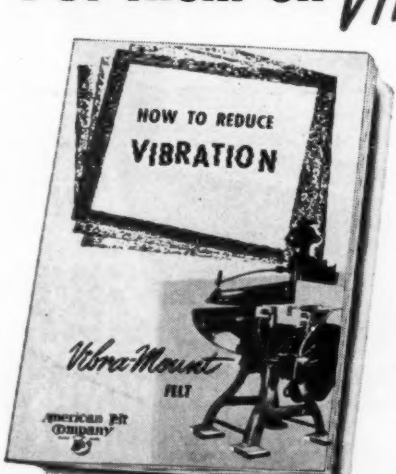

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ported that 95 per cent of service failures in machine parts are by fatigue. Thus the problem of fatigue is of utmost importance to the design engineer who must properly understand and recognize the phenomena involved in this type of failure in order to design equipment and machinery for successful operation.

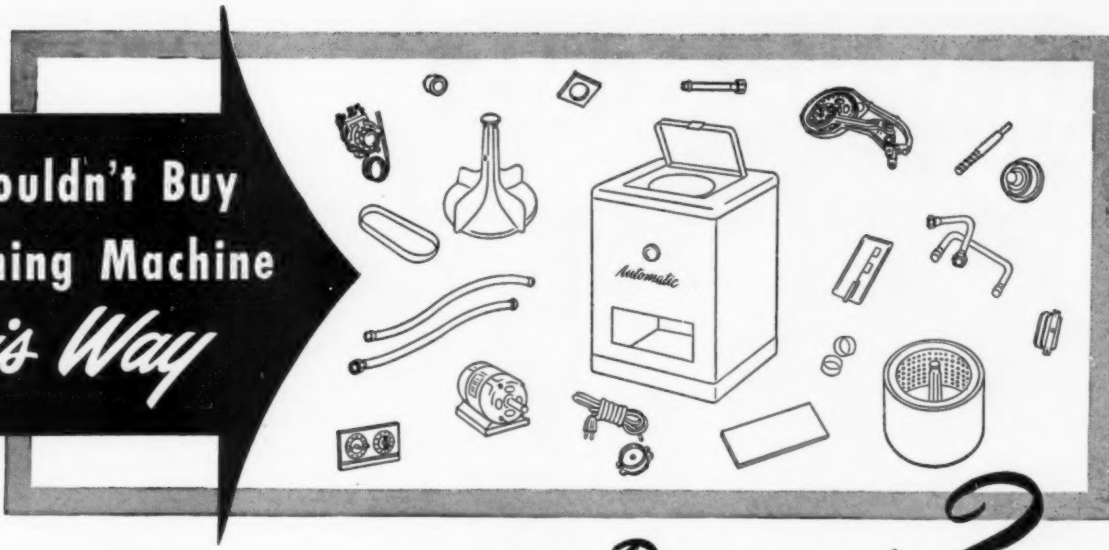
Data are Statistical

The fatigue of metals is of great scientific interest as well as practical importance. This peculiar type of failure occurs at stresses which are far less than those which produce fracture in the ordinary manner such as under a tensile load. This fact and many other fatigue phenomena require theoretical interpretation including a fundamental mechanism which explains how and why this type of failure occurs.

Fatigue is demonstrably a statistical phenomenon, but the fundamental factors which influence this behavior have not been understood. It has been known for some time that the fatigue life of a metal at a given stress varies statistically and more recently it was discovered that the endurance limit is also a statistical quantity and not an exact value. Thus both the fracture (finite life) and endurance ranges are subject to statistical variation and there is a finite probability of premature failures in the fracture range and of the occurrence of failure below the so-called endurance limit. The dependence of this behavior on metallurgical factors was not known and it was toward this objective that much of the experimental work was directed.

Sources of Error

Dispersion in fatigue data can arise from three principal sources: test machines, the preparation of specimens, and the metal itself. Poor alignment, calibration, and operation of the fatigue machines and lack of control of specimen preparation (surface finish, etc.) would obviously produce scatter in



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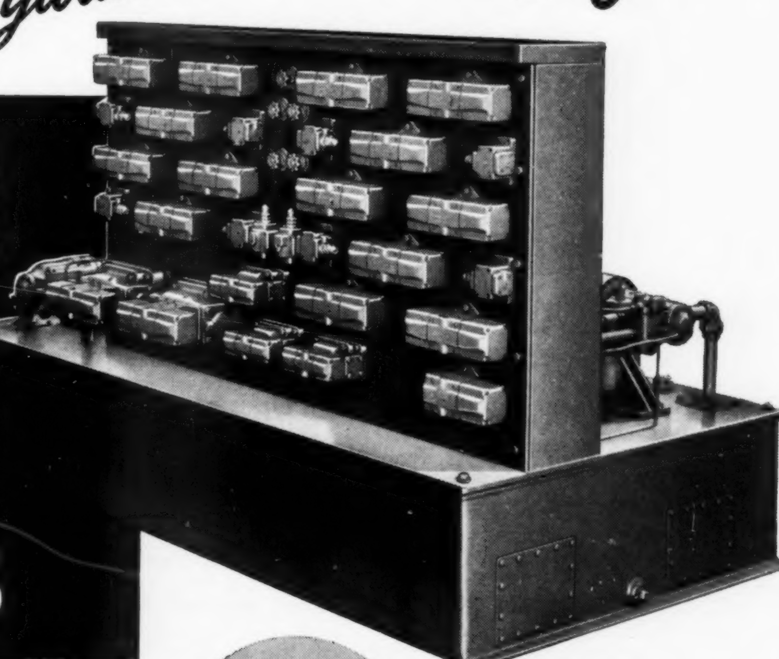

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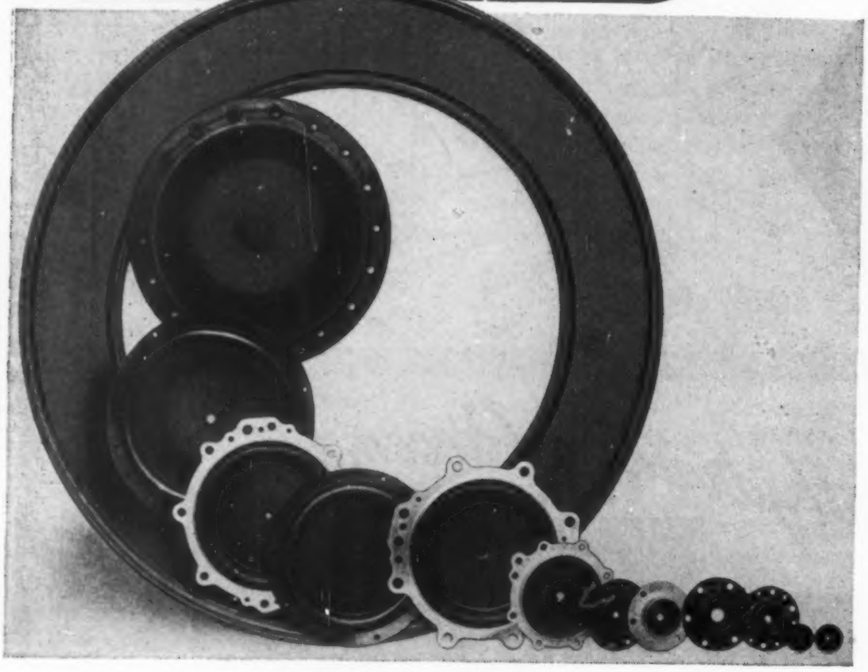
ENGINEERS AND BUILDERS OF OIL HYDRAULIC EQUIPMENT SINCE 1921



the test results. However, even when the experimental variables are properly controlled the fatigue life and endurance limit are statistical in nature, and this variability is inherent in the material.

It was thought that the statistical variation of fatigue properties depended upon the cleanliness, composition, strength level, and microstructure of the metal. For example, a material with a large number of inclusions might have a greater statistical variation in fatigue properties than a material of similar composition and microstructure with very few inclusions. Or, for a given composition and inclusion rating, the variability might depend upon the type of microstructure, and so on.

Statistics of the fatigue-fracture curves and endurance limits for a variety of materials were obtained from extensive fatigue tests and from a critical review of the literature. The results were analyzed to determine the relative effects of some metallurgical factors on the statistical nature of fatigue properties.

In addition, a number of related problems were studied including the dependence of statistical variations in fatigue life on stress level in the fracture range, location of crack initiation, size effect, understressing effect, and a form and possible method of plotting the *S-N* diagram. An approximate method of predicting premature failure in steel in the absence of statistical data was also determined.

As a result of this study and research, the following conclusions may be drawn:

1. Both the fatigue life and endurance limit are subject to marked variability and, therefore, these quantities cannot be stated as exact values, but must be represented statistically.
2. In view of the statistical nature of fatigue properties, the design engineer must, in choosing a factor of safety, recognize the fact that some failures can occur prematurely or even at stresses below the normally determined endurance limit.
3. The *S-N* diagram does not follow

the simple curve which is usually drawn but has a point of inflection and bends toward the stress axis at shorter life.

4. Statistical analysis of the measurements of the location of fatigue crack initiation provide further evidence of the inhomogeneous nature of the steel and its influence on the statistical behavior in fatigue.
5. The dispersion in fatigue life in rotating and vibrating tests increases with decrease in stress in the fracture range. In torsional fatigue, however, the dispersion is independent of the level of the stress.
6. The variability in fatigue properties depends upon the metal and is influenced by metallurgical factors.
7. The statistical variation in the fatigue life of nonferrous metals (e.g., copper and aluminum) is less than that for iron or steels, probably because the former materials have fewer inclusions and inhomogeneities.
8. Of the many factors which can influence the statistical behavior of fatigue properties, inclusions are the most important. Two steels of the same chemical composition (e.g., two heats of SAE 4340) but with widely different inclusion ratings have correspondingly different variabilities.
9. The effect of differences in composition and microstructure of iron and steels on the degree of dispersion is overshadowed by the inclusion contents of the materials. A steel of given composition can have greater or less variability than iron depending upon whether its inclusion rating is relatively high or low.
10. At a given inclusion rating, the material with higher ductility yields less scatter in the endurance limit, but the dispersion in fatigue life is essentially the same.
11. The understressing effect may, in part if not wholly, be interpreted as a statistical phenomenon. Experimental results conform to theoretical predictions based purely on selectivity.
12. The size effect can be explained on a statistical basis.

From NACA Technical Note 2719 (*June* 1952), *"Investigation of Statistical Nature of Fatigue Prop-*

erties." This comprehensive report contains 119 pages and is well illustrated with diagrams and photos.

Casting Aluminum and Magnesium Alloys

By W. D. Stewart

Staff Metallurgist
Central Metallurgical Dept.
Castings Div.
Aluminum Co. of America
Pittsburgh, Pa.

SUCCESSFUL application of aluminum and magnesium alloy sand and permanent-mold castings as engineering materials requires an understanding of the characteristics of these alloys and fabricating methods, the fundamental design considerations which are applicable to these alloys, and an appreciation of the effect of all of these factors on the characteristics and tolerances of the end product. Design tolerances may be considered from the standpoint of casting shape and dimensions as well as from the standpoint of service stresses and the physical properties of the part.

A fundamental problem important to the foundry in the production of a cast shape is the establishment of a sequence of solidification so that the change in unit volume on solidification can be compensated for satisfactorily. The designer should, wherever possible, use sections which are tapered so that they increase in thickness toward points accessible to feed metal. If tapered sections are not practicable, a uniform section thickness should be maintained. If it is necessary to use a design where light and heavy sections join, a gradual increase in thickness from the light to the heavy section is most desirable. This serves to minimize abrupt changes in solidification rate and promotes a condition more satisfactory for progressive solidification.

The low specific gravity of aluminum and magnesium alloys and the problems involved in pouring these alloys at high temperatures imposes a lower limit on the thickness of section which can be suc-

cessfully cast. This limitation applies also to the design of ribs, bosses, and fillets. In general, no radius on a cast surface should be less than the thickness of the thinner of the two joining sections. If the thickness of the larger of two sections exceeds the smaller by 50 per cent, a gradual blending of the lighter section to the heavier should be used in addition to the fillets.

When designing parts that are to be machined, the minimum machined radius even for small, lightly stressed castings should be 0.040-inch and all sharp corners should be broken. Whenever possible, this minimum radius should be doubled and best practice dictates the use of a machined radius equal to ⅛ of the thickness of the adjacent casting section. Care must be taken also to avoid stress concentrations at bolt holes, and a number of somewhat smaller uniformly spaced holes is preferred to one large hole. To provide sufficient strength in the threads where stud bolts are used in aluminum and magnesium alloy castings, a thread length equal to at least twice the diameter of the bolt should be used.

Inserts Resist Wear

It is often desirable to cast inserts of other metals in aluminum and magnesium castings in order to form wear resisting surfaces. Cast iron and steel are the most suitable materials, although copper and brass are sometimes used. Inserts are usually retained mechanically in the casting by knurling, grooving, or projections on the insert. Iron and steel inserts can also be metallurgically bonded to aluminum castings; for example, cast iron ring carriers are bonded to permanent-mold cast pistons for heavy-duty diesel engines.

Support of cores in castings sometimes requires core prints through a casting wall which may be objectionable from a design standpoint. In such cases, cores can be supported by chaplets, but this method is not advisable if the casting is required to be pressure-tight. A better method in this

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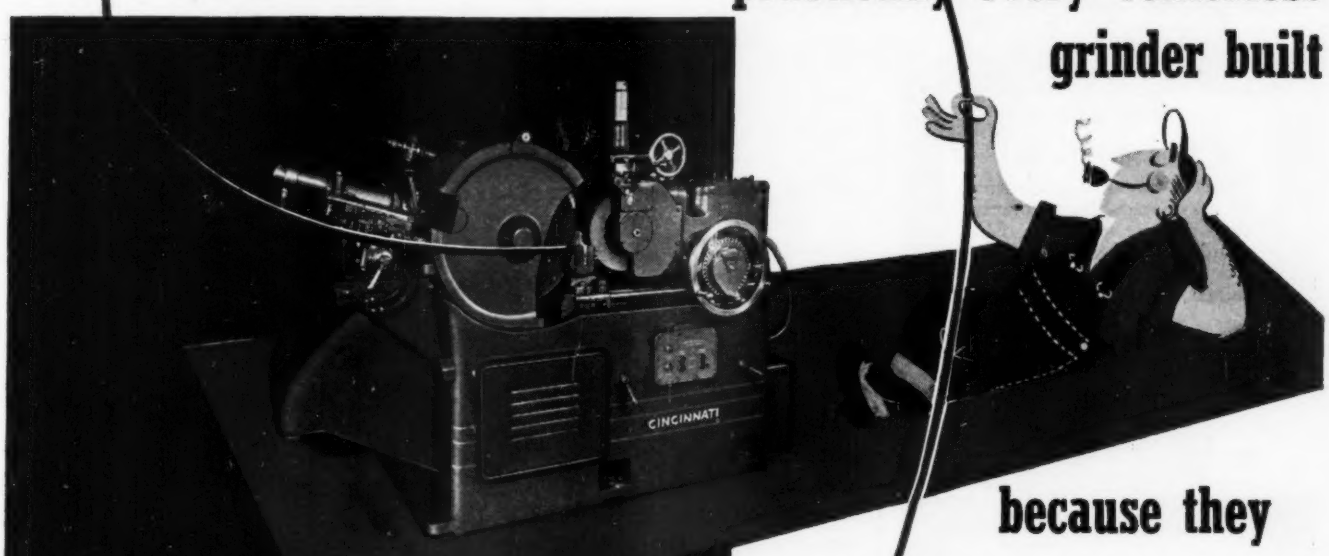

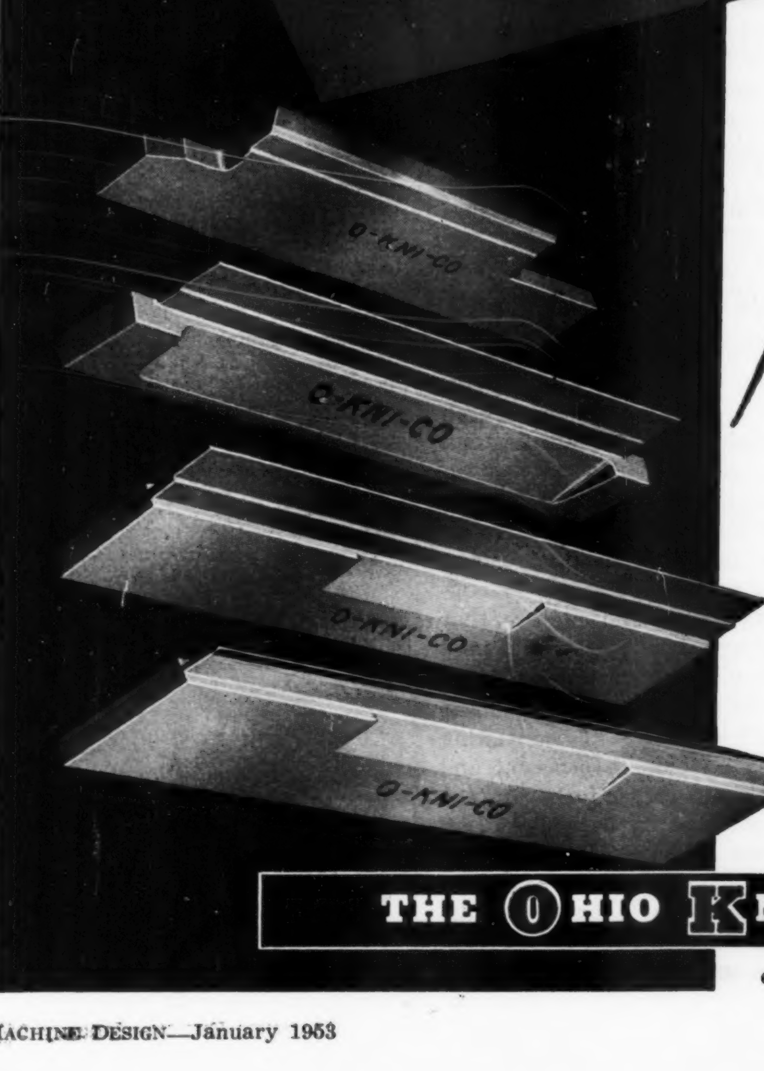

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Design Abstracts

case is to provide core support holes of adequate size, which can be sealed with welded, threaded or cupped plugs. Adequate beading to support such plugs is necessary.

Tests Can Mislead

Mechanical properties of aluminum and magnesium casting alloys for specification purposes are normally based on tests of individually cast specimens having a test section ½-inch in diameter. This test bar has been designed to provide optimum reproductibility of results and to reflect such changes in composition, metal handling practices, or heat treatment as may have occurred in its production. However, the properties of separately cast test bars do not necessarily represent the properties of the actual casting. These properties may be higher or lower depending on a number of factors which influence the solidification rate of the metal in the mold or the soundness of the casting section. For example, test specimens machined from a heavy section of a casting may provide lower properties than separately cast specimens because the rate of solidification is relatively slow. Conversely, the high solidification rate of a thin section may result in higher mechanical properties than those shown by separately cast specimens.

This condition is not peculiar to aluminum and magnesium alloys, but is true to various degrees of all cast metals. Because of this factor, the properties of test specimens machined from a single casting will also vary depending upon the location in the casting. Many engineering specifications recognize this factor and require that the average tensile strength and elongation of specimens machined from the heaviest, intermediate, and lightest section of the casting be not less than 75 per cent and 25 per cent, respectively, of the values required for separately cast specimens. There is no general rule by which this relationship can be determined in a particular design, and the designer, through experience, must develop the proper

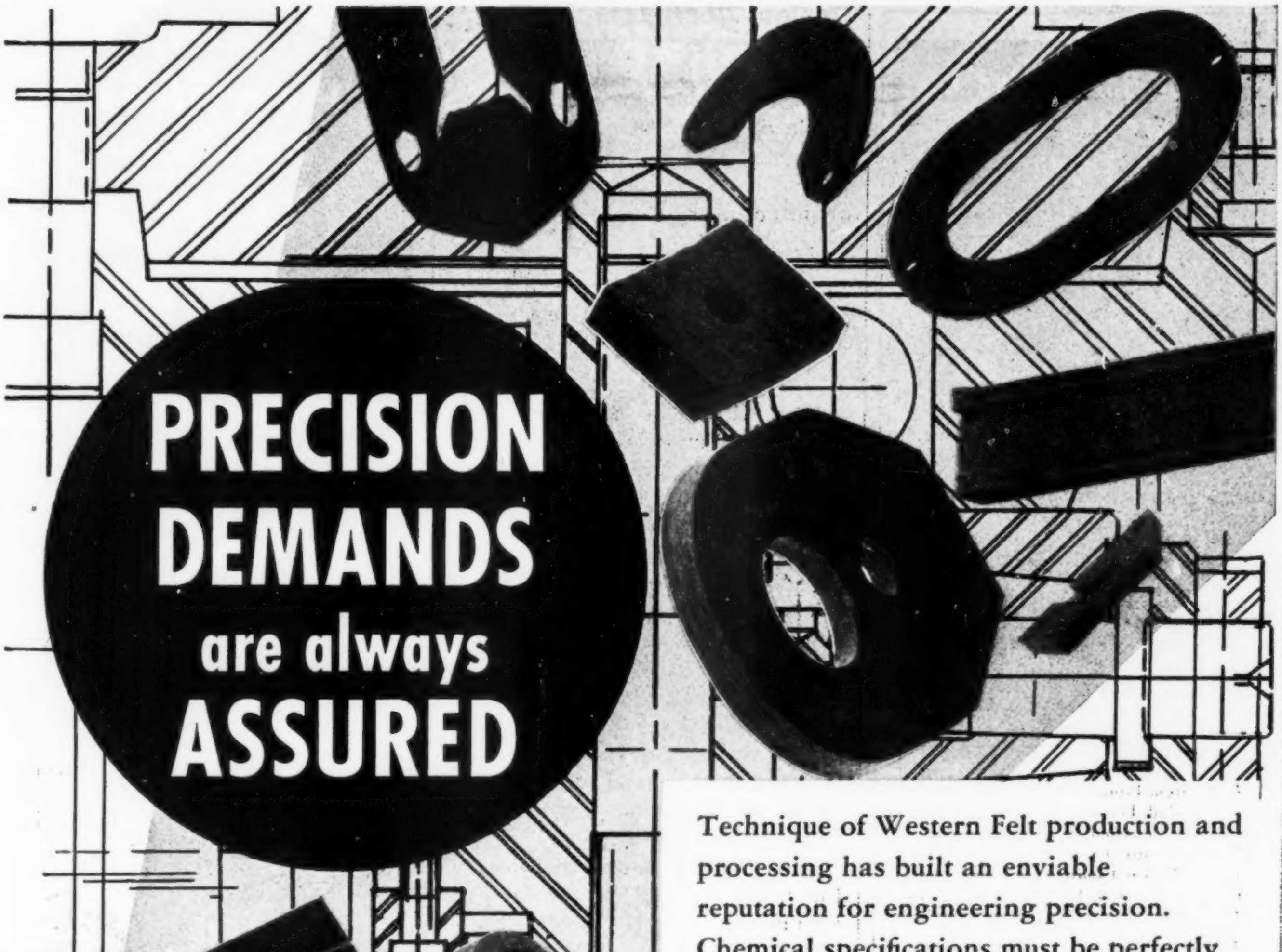
factor of safety to apply to the mechanical properties specified for an alloy in evaluating the design stress to be applied.

Sand Castings: The sand casting process is the most versatile method of producing a cast shape and is, in fact, characterized by its universal adaptability. This process is employed for the production of (1) small quantities of identical castings, (2) parts requiring intricate coring, (3) large castings, and (4) castings of a design suitable to modern high-speed mold equipment and methods where production costs lower than those of other fabricating methods can be established.

Type of pattern equipment selected will depend largely upon the dimensional accuracy required and the total quantity of castings required or the production rate which is desired for control of unit cost. Shrinkage allowance for both aluminum and magnesium alloy sand castings may be as much as 5/32-inch per foot, but variations in casting design, alloy, and resistance to contraction offered by the mold may make necessary an allowance as small as 1/10-inch per foot. The actual shrinkage allowance for a particular casting design must be determined by experience.

Molding in sand is not a precision operation because of variations inherent in the process. These variations are introduced by, among other factors, ramming of the sand onto the pattern, drawing the pattern, and setting of cores. Variations in the dimensions of the resulting casting are introduced by trimming and heat treating.

The overall effect of these fabrication tolerances is reflected in the dimensional tolerances that can be maintained in the sand casting process. The standard tolerance for both aluminum and magnesium sand castings is $\pm 1/32$-inch. This tolerance will apply to such dimensions as length, flatness, and wall thickness in most castings made from mounted pattern equipment having a maximum dimension of 12 inches. Beyond this point, it is inaccurate to generalize about sand casting tolerances because of the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effect of casting size. As an example of the effect of size on the tolerance for length dimensions, the following can be considered examples of acceptable tolerances:

Length Dimension (inches)	Tolerance (inches)
Up to 12	$\pm\frac{1}{32}$
12 to 24	$\pm\frac{3}{64}$
24 to 36	$\pm\frac{1}{16}$
36 to 60	$\pm\frac{3}{32}$
Over 60	$\pm\frac{1}{8}$

Wall thickness tolerances are also a function of casting size, and the size and weight of cores which must be set and supported in the mold. For example, in large aluminum-alloy sand castings weighing 400 to 1000 pounds, a wall thickness tolerance of $\pm 3/32$-inch may sometimes be necessary.

All the tolerances mentioned so far are for castings made in green sand molds with or without dry sand cores. When the entire mold is made in dry sand, closer tolerances can be held provided the casting is not too large.

The physical characteristics of the sand, the quality of the pattern surface, and the molding technique all govern the surface smoothness of sand castings. Although a relatively smooth surface can be obtained on aluminum alloy castings with fine sand, the use of such sand is limited to rather small castings. Larger castings require somewhat coarser sand in order to provide the increased mold permeability necessary for better casting soundness. This latter factor is most important in the production of sound magnesium alloy castings, and, in general, aluminum sand castings have somewhat better surface smoothness than magnesium castings. Special requirements for surface smoothness on sand castings exert a marked influence on their cost, and the designer should carefully consider the necessity for specifying excessively smooth as-cast surfaces.

Permanent-Mold Castings: In the permanent-mold process, metal molds and cores are used and the metal is poured into the mold cavity under a normal gravity head. The semipermanent-mold process is a variation that offers somewhat more versatility through the use

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MACHINE DESIGN

of dry sand cores, thus overcoming many of the design limitations imposed by metal cores. Permanent-mold castings are characterized by their metallurgical superiority, closer dimensional tolerances, superior surface finish, pressuretightness, and for certain sizes or designs, greater speed of production and lower cost per piece than sand castings.

The metallurgical superiority of this process is more pronounced in the case of aluminum castings than with magnesium castings. This superiority is due to the chilling imposed by the metal mold and the consequent refinement of the structure and reduction in porosity in the castings. This results in a substantial improvement in the strength over similar parts cast in sand. Magnesium alloys will, in most cases, show somewhat higher mechanical properties when cast in permanent molds.

Minimum Dimensions

As with sand castings, it is difficult to define the minimum section thickness that can be produced by the permanent mold process because of the effect of casting size upon this factor. A section thickness of 1/8-inch over an area of 3 inches or less is usually considered to be the minimum acceptable. For larger areas, the minimum section thickness should be 5/32-inch and a minimum section of 3/16-inch is usually required if the shortest dimension of the web area exceeds 6 inches. The minimum diameter for cored holes is dependent upon a number of factors including depth and location with respect to various parts of the mold. In general, best design practices would be to provide for no cored holes less than 3/8-inch in diameter. Smaller holes can usually be produced more economically during machining.

Dimensional tolerances depend upon the size and complexity of the casting and the location of the dimensions involved with respect to the moving parts of the mold. Tolerances lower than those possible with sand castings are generally provided, ± 0.020-inch being commonly maintained for small cast-

Design Abstracts

ings. Tolerances are usually given as $\pm 1/64$-inch for the first 1 inch or less, plus 0.001 or 0.002-inch per inch of added dimension depending on whether the dimension is affected by the moving parts of the mold. Draft allowances should be as generous as possible with 1 degree considered minimum for outside draft and 3 degrees more desirable. In recesses in the casting formed by the mold, a 2-degree draft is considered minimum with 5 degrees desirable. For cores the minimum draft is ½-degree with 2 degrees desirable. The selection by the designer of the desirable degree of draft, wherever possible, has a significant effect upon the cost of the casting because of the influence of this factor upon the production rate which can be expected from the mold.

As in the case of sand castings, it is possible to maintain closer than normal tolerances on certain specific dimensions. Examples are permanent-mold pistons, where diameters across chucking points are held to ± 0.010-inch and head thicknesses are held to ± 0.010-inch.

Machine allowances for permanent-mold castings can be somewhat less than those indicated for sand castings. For castings up to 10 inches long, a minimum machining allowance of 1/32-inch should be provided, with 3/64-inch desirable. For castings over 10 inches long, the minimum machining allowance is 3/64-inch with 1/16-inch desirable. In semipermanent-mold castings on surfaces formed by sand cores, 1/16-inch minimum machining allowance should be provided.

The surface finish of permanent-mold castings is governed by the smoothness of the mold surfaces and the proper control of mold coatings, and is superior to that provided by sand castings. Surfaces produced by sand cores in semipermanent-mold castings are, of course, dependent upon the smoothness of the core.

From a paper entitled "Tolerances and Specifications for Aluminum and Magnesium Castings" presented at the ASME Fall Meeting in Chicago, Ill., September 1952.

New Machines

Materials Handling

Electric Fork Truck: Model FS-25, for loads up to 2500 lb and 48 in. long. Features worm-gear driven power axle; gyroscopic trailing axle construction—truck chassis remains level even when all four wheels are on different surface levels; dynamic braking to prevent motor burnouts caused by sudden reversals of direction; powerful motor which will handle extreme overloads. Has standard telescoping lift with 130½-in. travel. Overall length, 119 in.; width, 34 in.; turning radius, 65½ in.; speed, up to 7 mph. *The Baker Raulang Co., Baker Industrial Truck Div., Cleveland, O.*

Furnace Charger Truck: Model A-3481 has capacity of 4000 lb. Load pickup and deposit is accomplished through tilting action of load carrier, which pivots backward a maximum of 10 degrees. Chassis is equipped with gas-electric power source. Standard battery power source recommended is one of either 30 or 36-v type and 450-amp-hr rating. *The Mercury Mfg. Co., Chicago, Ill.*

Metalworking

Broaching Machine: Hyspeed 20-in. unit suited to high production broaching of specialized medium and small jobs, such as splines, wrench slots, internal wrench hexes, etc. Machine cycle is started by foot pedal which can be operated from either side. Hydraulically operated and electrically controlled; electric control panel has selector switch which determines an automatic return stroke or the conventional broaching stroke which stops at the end of each cutting stroke. Integral coolant system pumps either light or heavy cutting solvents at considerable volume. Maximum pull, 6000 lb; maximum feed stroke, variable between 2 and 27 fpm; full return stroke of 20 in. requires 1½ sec-

onds. Cycles at 700 strokes per hour. Floor space required, 3 by 9 ft. *Zagar Tool Inc., Cleveland, O.*

Flash Trimmer: Designed for use in a continuous process rolling or pickling line. Placed in the line adjacent to the flash welder. Welded sheet is moved to approximate center of trimmer, which is provided with hydraulic horizontal traverse 6 in. each way from center for final alignment of machine with the weld. Operator stops weld near center of machine; traverses machine to final alignment. Pushbutton controls lowering of upper housing and clamping of the sheet. Rams move automatically and simultaneously across the sheet, removing the flash. At the end of the fixed travel, limit switches unclamp and raise upper housing, rams return to front of machine, and machine returns to its central position of traverse. Will trim materials from 0.056 to 0.250-in. thick. Has 3-in. vertical movement and clearance between dies. *Morton Mfg. Co., Muskegon Heights, Mich.*

Multiple Head Grinder: For production grinding of solid carbide blanks. Produces finish of 1 to 2.5 rms and gage block flatness using 150 grit diamond wheel. Holds dimensions to ± 0.0002-in. Wheel can be dressed and trued to 0.0005-in. in a few minutes. Long wheel life; diamond salvage is less than one carat per pound of sludge. *Spike Mfg. Co., Farmington, Mich.*

Portable Nibbler: Cuts through 14 gage stainless steel, C. R. steel, galvanized iron or softer materials in proportion without distortion on either side. Also cuts holes in tubes and ducts without damaging original contour. Can be used as a hand tool or mounted in a vise for bench operations. Minimum cutting radius, 7/8-in. Weighs 7½ lb; is 10 in. long. *Fenway Machine Sales Co. Inc., Philadelphia, Pa.*

Punch Presses: Two power-operated models each having rated capacity of 5 tons. Will punch 4-in. diameter hole in 16 gage (0.062-in.) sheet steel or a 3/8-in. hole in 3/16-in. thick steel plate. Feature deep throats; rate of production is 180 strokes per minute. Operated by foot control. Moving parts

housed in steel cabinet. Require floor space of 17 by 28 in.; need not be bolted down. Driven by ½-hp motor, either single-phase 110-220 v or three-phase 220-440 v ac. *O'Neil-Irwin Mfg. Co., Lake City, Minn.*

Arc Welder: Model WD-42AGW, 200-amp engine-driven dc welder with 60 per cent duty cycle. Has current range of 40 to 250 amps; can be used with a variety of electrode sizes. Consists basically of GE type WD42 generator and Wisconsin air-cooled engine. Designed to fit crosswise in standard pickup truck. Forced ventilation keeps internal temperature within safety limits when welder is operated on 60 per cent duty cycle at rated load. Vacuum type device idles engine when welding is not being done. *General Electric Co., Schenectady, N. Y.*

Horn Press: Capacity, 40 tons. Features heavy duty frame and easy adjustment. Available either plain or geared. Specifications include: diameter of shaft at main bearings, 3½ in.; table area of bolster, 18 by 24 in.; opening in bed, 10 by 14 in.; horn hole and length, 6 by 19 in.; die height, table to ram, 6 to 18 in.; die height, center of horn hole, 8 in.; bolster thickness, 2 in.; stroke, standard to maximum, 3 to 6 in.; ram adjustment, 2 in.; ram area, 7 by 8 in.; strokes per minute, 50 to 100. *Diamond Machine Tool Co., Pico, Calif.*

Belt Sander: For fine burring, grinding and polishing operations on metals, rubber, plastics, glass, etc. Belt wheels, 4 inches in diameter, are mounted on heavy duty, wide, double-row double-sealed ball bearings. Unique trackage adjustment ensures positive wheel alignment. Top wheel supporting arm is rocker mounted and pivoted for belt takeup. A spring load ensures correct belt tension when using the back-up plate. Requires no auxiliary belt trackage devices. Pivoted back-up unit retains parallelism with the belt at any angular setting of table. Designed to operate at any surface speed up to 8000 fpm; requires ¼-hp, 1750 rpm motor. Accommodates any standard belt width

from ¼ to 1 in. by 44 in. long. *Benchmaster Mfg. Co., Gardena, Calif.*

Thread and Worm Cutter: For installation on a standard lathe. Actual cutting is done by a tool which moves around workpiece eccentrically at high speed, in a single continuous operation. Cuts long standard vee threads, acme, buttress and rounded threads and, within close limits, modified square threads. It is claimed that threads cut with this equipment are more exact in pitch than those produced under similar machining conditions by twining. There are no pitch errors produced by the torsion of a workpiece. High-speed cutting eliminates tearing of flanks and "smearing" of workpiece. Velocities are between 800 and 2000 fpm, depending on material. *National Threading Machine Co., Paterson, N. J.*

Power Hack Saw: No. 5 Hy-Duty Keller machine has capacity for a 9¼-in. round or an 8 by 9-in. flat. Lifts automatically on reverse stroke. Variable power pressure regulator for the blade provides pressure from 0 to 200 lb. Cuts any material from lightest wall tubing to heavy shafting. Equipped with swivel base vise with quick-adjustable sliding jaws and single screw. *Sales Service Machine Tool Co., St. Paul, Minn.*

Welding Torch: Water-cooled model HW-10 Heliarc torch for inert gas-shielded arc welding. Welds almost all commercial metals up to approximately ⅛-in. thick. Has rated capacity for continuous duty of 300 amps. Does not overheat even if used for long periods at full rated current. Cooling water flows directly into torch body and is circulated to torch head, through the water jacket, back through the head again and out through the power cable. This water cooling system makes possible placing the collet body closer to the arc without danger of damage from excess heat. Electrodes can be changed or adjusted in a few seconds' time without using a wrench. A quarter of a turn releases torch cap, and steel collets align new electrodes automatically. Collets are available for 0.040, 1/16,

New Machines

3/32 and 1/8-in. diameter electrodes. Two torch caps are available, permitting the use of either 3 or 7-in. electrodes. *Linde Air Products Co., New York, N. Y.*

Pipe and Tube Notcher: Twin-Notch Arc-Fit for use with a punch press. Notches both sides of pipe end with a single stroke of the press. Requires less than three seconds' time. Tubing is fed into die and die punch enters tube end. As press ram comes down, punch is driven laterally to notch one side of tube end, then immediately in the opposite direction to notch the other side. Cutting is from the inside out; clean edges require no further finishing. Alignment of notches is automatic. Interchangeable dies, punches and spacers make possible notching 1/2 to 2-in. pipe and tubing. *Vogel Tool and Die Corp., Melrose Park, Ill.*

AC Welding Machines: New models in 300, 400 and 500-amp capacity replace previous 300 and 500-amp machines and incorporate new features and mechanical changes. Have "arc booster" circuit. Geared crank facilitates setting of amperage. Electrical and mechanical control parts are accessible without removing case. Reconnection of input circuit for 220 or 440 volts can be accomplished without replacing any coils. Improved downdraft ventilation system brings air in from top of case, circulates it by means of baffles through windings and prevents any air from being recirculated. Large lift bale on top of case accommodates standard crane hook. *The Lincoln Electric Co., Cleveland, O.*

Plant Equipment

Steam Cleaner: Improved Speedylectric Model JC-10 features operating pressures to 150 psi and closer fingertip control of steam-detergent mixtures. Uses steam from built-in, high-pressure boiler. High velocity jet of hot dry steam and solvents is applied as needed. Unit is free of low water danger—the boiler water itself is the electric resistance heating element—and if there is no water, no current passes and no steam is generated. Available for single or polyphase

power supply of 220, 440 or 550 volts, ac. Requires 26 by 17 in. floor space; weighs under 200 lb. *Livingstone Engineering Co., Worcester, Mass.*

Measuring Machine: For measuring the length of electric wire, cable, flat strip steel, tape, cordage, small hose, plastic products and other materials. Measuring wheel rides on top of the material. Brake stops wheel when material runs out. Counter is four-digit reset type; direct reading on one dial; will record up to 400 counts per minute. Model No. 0 measures up to and including ¾-in. diameter material; No. 1 measures up to and including 1¼ in. diameter material. *Neal Co., Omaha, Neb.*

Hand Engraving Tool: Recipro-Tool engraves letters, numbers or identification marks on any material of any hardness or with any surface finish. Can also be used for riveting, filing, burring, peening or cutting glass, gaskets, plastics, shim stock and other materials. Delivers 5¼-lb impact, 7200 strokes per minute which are adjustable from 0 to ⅛-in. Positive locking device maintains length of stroke accurately until reset. Available with a variety of hardened alloy-steel bits for materials up to Rockwell 50C or with a diamond-tipped bit. Features built-in lubrication, low center of gravity, finger grip close to chuck, fingertip switch control and electric cord leading under operator's hand. *S. C. & L. Machine Co., Rockford, Ill.*

Processing

Heat Treating Furnaces: Three electric high-speed tool steel models have operating temperatures ranging from 1650 to 2500 F. Chamber sizes of 6¼ by 4 by 9, 12 by 8 by 18 and 12 by 8 by 24 in. are available. Power is controlled by multiple-tap transformers arranged for select voltages to be applied to the heat elements located at top and bottom of the chamber. Tap changing switches and ammeters are placed at the front of the furnace for operator's convenience. Temperature is controlled by means of a standard

Pyrometer arrangement. *Cooley Electric Mfg. Corp., Indianapolis, Ind.*

Wet Blasting Machine: Model 64 Liquamatte for precision cleaning and finishing of large and heavy pieces. Typical manufacturing and maintenance applications include forging, die casting and drawing dies; glass, plastic and rubber molds; surfacing tools, such as long broaches before and after plating; blending different types of finishes on aluminum; removing heat treat scale, and deburring. Will handle single pieces or loads up to 3500 lb. Has two 49 by 36-in. counterbalanced doors and a 6-ft long accessory tank to receive parts from blast chamber for rinsing. Can accommodate two operators at one time. Three individually controlled abrasive nozzles have different sized jets. Pushbutton controls are located on one panel near work station. Has vertical pump for slurry agitation and recirculation which is adaptable to rugged service and eliminates all suction piping, valves and fittings. *American Wheelabrator & Equipment Corp., Mishawaka, Ind.*

Testing and Inspection

Profile Projector: For use in automotive, aircraft, ball bearing and cutting tool industries. 200-in. unit features two sets of precision optics and a horizontal screen of 60-in. diameter. Tracings or templates can be mounted on viewing table. Optics will provide 20 to 100 times magnification for work up to 3-in. diameter; for work up to 6 in., 10 times magnification is available. *Engis Equipment Co., Chicago, Ill.*

G-Accelerator: Model C, 8-ft diameter. Incorporates combined optical, pneumatic and electrical systems for testing aircraft and guided missile components. Object to be tested is secured to either of two symmetrical mounting platforms on the rotating arm; arm is balanced by securing similar mass on opposite end. Electrical leads and air connections are carried to the mounting platform through slip rings and a rotary air gland. Test

object can be seen as though stationary through a telescopic optical system which permits close observation of the part at any time during testing. Indicating tachometer and precision timer, combined with a solenoid operated rpm counter, measure speed accurately to approximately 0.1 per cent, depending upon time interval. Controlled speeds, adjustable from 5 to 280 rpm and for developing normal acceleration forces up to 75*g*, are attainable. Machine returns to preset speed when starting switch is actuated. Time required to reach full speed from standstill is approximately 30 seconds. Maximum size of test object limited to 18 by 18 by 24-in. volume weighing up to 100 lb. Maximum acceleration force is limited to 2000*g*-lb (100 lb at 20*g*, 40 lb at 50*g*). Size, 96-in. diameter, 24 in. high, 31 in. wide with side sections removed. *Genisco Inc., Los Angeles, Calif.*

Gear Case Tester: Gear cases and related mechanical power transmission components may be tested under full load and full speed by "square rig" method, whereby power of driving motor is only 5 to 10 per cent of rated power of unit to be tested. Unit works without supervision and stops automatically in case of malfunctioning of any part. *Technical Development Co., Philadelphia, Pa.*

Magnifier: Electrically lighted, designed for detailed examination of materials and products through self-contained direct illumination on the object. Lens magnifies five times and is highly corrected to eliminate spherical and chromatic aberration and distortion. Has wide, flat field. Equipped with either of two types of illuminator handles, one battery-powered, the other for connection with a 110-v power source. *Bausch & Lomb Optical Co., Rochester, N. Y.*

Hardness Tester: Capacity, 3-in. Features several improvements. A and D scales have been added to C-B and E scales, making possible the testing of very thin sheet steel. Has visual means of setting to different load factors for the different scales used, thus preventing mistakes in setting. *J. P. Newman Co., Alhambra, Calif.*